Iron and Steel Engineer

Established 1907 as Proceedings of A. I. & S. E. E.

J. F. KELLY, EDITOR

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ASSOCIATION OF IRON AND STEEL ELECTRICAL ENGINEERS

EXTEND A CORDIAL INVITATION

TO ATTEND THEIR

28th CONVENTION

AND

IRON AND STEEL EXPOSITION

JUNE 20-21-22-23, 1932

WILLIAM PENN HOTEL

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TECHNICAL SESSIONS 9:30 A. M.

Electrical Engineering Division
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Mr. J. J. Booth, Electrical Superintendent, National Tube Co., Gary, Indiana, President of the Association of Iron & Steel Electrical Engineers for the year 1931-1932.

The A. I. & S. E. E.— A Factor in the Iron and Steel Industry

▲

By J. J. BOOTH

President of A. I. & S. E. E.

The Iron and Steel Industry of our country is at the present time, and has for the past number of months, been experiencing difficulties which have been heretofore unknown, and up to the present time have by no means been solved in their entirety. These problems, which have been brought on and accentuated by the general economic condition of our country, cover practically every phase of the Iron and Steel Industry, in production, sales, and finance.

The two major problems confronting the steel mill executives of today comprise keeping the payroll intact by distributing equitably as nearly as might be, among the workers the available work, and conserving in so far as practicable the cash surplus.

Between these two major questions we may list in part the following operating and producing problems:

Small demand and insufficient orders to keep mills operating near normal production.

Difficulties encountered in the operation of coke ovens, resulting in long coking hours and inefficient production and handling of by-products.

Curtailed production of blast furnaces necessitating blowing out of furnaces, banking, or operating on reduced pressure.

Intermittent operation of open hearth furnaces, accompanied by abnormal expansion and contraction of flues, checkers, etc.

Frequent heating and cooling of soaking pits, continuous and reheating furnaces.

Periodical starting and stopping of rolling and finishing mills accompanied by many roll changes.

Poor load factor on power houses combined with the use of emergency fuel.

The above problems plus many more, which space will not permit enumerating, are the causes of exceedingly high operating costs, accompanied by inefficient use of fuel, extremely high maintenance, rapid depreciation of buildings and machinery, low practice, excess scrap, and in many instances inferior quality product.

The above problems which face our industry are so intricate and so difficult of solution, that no one individual or no one group of men will be able to solve them alone. It will require the individual effort and the co-operation of each executive and his organization together with our government, bankers, etc. to solve the many economic problems, thereby improving our business and financial structure and bringing about the upturn of business.

Despite the present unrest and unsettled economic condition, we have implicit faith in our country, its economic structure, and the sound judgment of its people. We believe that in due time we will overcome all obstacles, and that prosperity will be here again, with full time employment, and that the present conditions will soon be forgotten.

The Association of Iron & Steel Electrical Engineers has for its basic purpose a broad vision not only for its own interests, but for the interests of all in any way concerned with the Iron and Steel Industry and Allied Arts. The Association studies, investigates, and makes research into those factors which may help the growth, prosperity, and welfare of the Iron and Steel Industry. We are, therefore, now more anxious than ever to do our part and assume our share of responsibility along the several lines of engineering and research to assist in the return of business and normal production.

The Association of Iron & Steel Electrical Engineers offers, in its 1932 Convention and Exposition, one of the best balanced programs in its history. This program is both technical and practical, backed up by years of technical training and practical experience.

We cordially invite all to attend this Convention and Exposition and not only gain for yourself increased knowledge, but exchange ideas with your fellow engineers whose problems may be similar to your problems.

Monday, June 20, 1932

9:00 A. M. Registration: Chairman, F. D. Egan, Elec. Supt., Bethlehem Steel Co., Lackawanna, N. Y.

9:30 A. M. Business Session, Italian Room. Chairman, J. J. Booth, President A. I. & S. E. E., Supt., Electrical Dept., National Tube Co., Gary, Ind. Secretary, Wm. Jackson, Elec. Supt., Carnegie Steel Co., Munhall, Pa.

Report of Committees.

Technical Sessions, Italian Room. Chairman, J. J. Booth, Supt., Electrical Dept., National Tube Co., Gary, Ind. Secretary, Wm. Jackson, Elec. Supt., Carnegie Steel Co., Munhall, Pa.

10:00 A. M. "Electrical Developments in the Iron and Steel Industry" by W. H. Burr, Elec. Supt., Lukens Steel Company, Coatesville, Pa.

10:30 A. M. "Progress in the Electrical Welding Science" by G. E. Doan, Associate Professor, Metallurgical Engineering Department, Lehigh University, Bethlehem, Pa.

11:00 A. M. "Substitution of Fabricated Steel for Castings" by W. E. Watson, Youngstown Welding Company, Youngstown, Ohio.

11:30 A. M. "Deseaming of Slabs" by J. F. Lincoln, President, The Lincoln Electric Co., Cleveland, Ohio.

Safety Engineering Division, Technical Session, Italian Room. Chairman, J. F. Hunter, Safety Engr., Carnegie Steel Co., Pittsburgh, Pa. Secretary, E. F. Blank, Director of Safety, Jones & Laughlin Steel Corp., Pittsburgh, Pa.

1:30 P. M. "Safety Regulations Necessary When Working on High Tension Lines" by A. I. & S. E. E. Safety Engineering Division.

Discussion: By Public Utility Companies and Steel and Industrial Plants.

7:00 P. M. Official opening of the Iron and Steel Exposition — William Penn Hotel.

9:00 P. M. Informal Reception for Members, Exhibitors and Guests—Main Ball Room, William Penn Hotel. Chairman, W. E. Miller, Elec. Supt., Bethlehem Steel Co., Johnstown, Pa.

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OFFICIAL
CONVENTION AND IRON
ausp
ASSOCIATION OF IRON AND
JUNE 20, 21,
WILLIAM PENN HOTEL

Tuesday, June 21, 1932

Technical Session, Electrical Engineering Division Italian Room, William Penn Hotel.

Chairman, F. O. Schnure, Elec. Supt., Bethlehem Steel Co., Sparrows Point, Md. Secretary, J. I. Kaspari, Asst. Elec. Engr., Weirton Steel Co., Weirton, W. Va.

9:30 A. M. "Standard Control Apparatus" by E. L. Upp, Elec. Supt., National Tube Company, McKeesport, Pa.

10:00 A. M. "Ladle Crane Practice in the Iron and Steel Industry" by F. W. Cramer, Chief Elec. Engr., Republic Steel Corp., Youngstown, Ohio.

10:45 A. M. "Electron Tube Applications in Steel Mills" by T. R. Rhea, General Electric Co., Schenectady, N. Y.

"Devices Actuated by Light" by A. F. Corby, Jr., Weston Electrical Instrument Corp., Philadelphia, Pa.

"Profitable Uses for Electronic Devices Responsive to Light in the Iron and Steel Industry" by W. B. Shirk, Steel Mill Engr., Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa.

Wednesday, June 22, 1932

Technical Session, Electrical Engineering Division Italian Room, William Penn Hotel.

Chairman, J. D. Donovan, Elec. and Mech. Supt., Republic Steel Corp., Massillon, Ohio. Secretary, J. A. Morgan, Elec. Supt., Carnegie Steel Co., Braddock, Pa.

9:30 A. M. "Design and Performance of a Fully Compensated DC Machine and What it Means to the Steel Industry" by M. E. Thompson, Consulting Engr., and I. S. Nippes, Elec. Engr., Elliott Company, Ridgway, Pa.

10:30 A. M. "Determination of Motor Capacity for Main Roll Drives" by H. A. Winne, Industrial Engineering Department, General Electric Company, Schenectady, N. Y.

PROGRAM

AND STEEL EXPOSITION
ices
STEEL ELECTRICAL ENGINEERS
22, 23, 1932
- PITTSBURGH, PA.

Technical Session, Combustion Engineering Division, Cardinal Room, William Penn Hotel

Chairman, G. T. Hollett, Comb. Engr., Illinois Steel Co., South Chicago, Ill. Secretary, T. J. Ess, Comb. Engr., Republic Steel Corp., Massillon, Ohio.

9:45 A. M. "Water Cooled Furnaces for Steam Boilers" by H. J. Kerr and R. M. Hardgrove, Babcock & Wilcox Company, New York, N. Y.

10:15 A. M. "Development of Air Cooled and Insulated Refractory Walls" by R. C. Denny, Consulting Engr., and R. D. Foltz, Pres., M. H. Detrick Company, Chicago, Ill.

11:00 A. M. "The Influence of Atmosphere and Temperature on the Behavior of Steel with Respect to Scale Losses" by Dr. D. W. Murphy, University of Michigan, Ann Arbor, Mich.

2:00 P. M. "Maintenance and Care of Recording Instruments Used in the Steel Industry" by Martin J. Conway, Fuel Engr., Lukens Steel Company, Coatesville, Pa., and R. C. Kennan, Service Engr., Brown Instrument Company, Philadelphia, Pa.

2:45 P. M. "Fuel Economy Under Curtailed Operating Conditions" by H. Dobrin, Consulting Combustion Engineer, Pittsburgh, Pa.

3:15 P. M. "Higher Steam Pressures in the Blast Furnace—Steel Mill Power Plants" by Charles W. E. Clarke and Walter P. Gavit, United Engineers and Constructors, Inc., Philadelphia, Pa.

Thursday, June 23, 1932

Technical Session, Mechanical Power Transmission Division, Italian Room, William Penn Hotel

Chairman, J. H. Van Campen, Chief Engineer of Youngstown District, Republic Steel Corp., Youngstown, Ohio. Secretary, L. F. Coffin, General Master Mechanic, Bethlehem Steel Company, Sparrows Point, Md.

9:30 A. M. "Gear Transmission."

Open discussion by Mechanical Engineers, Electrical Engineers of the Iron and Steel Industry, Mill and Gear Builders of the United States.

Technical Session, Lubrication Engineering Division, Italian Room, William Penn Hotel

Chairman, C. C. Pecu, Lubrication Engineer, Bethlehem Steel Co., Lackawanna, N. Y. Secretary, Frank Gray, Testing and Lubrication Engineer, Illinois Steel Co., Gary, Ind.

10:30 A. M. "Lubrication and Lubricants for Roll Necks" by Walter D. Hodson, President, The Hodson Corporation, Chicago, Ill.

11:30 A. M. "Micarta Roll Neck Bearings" by W. B. Shirk, Steel Mill Engineer, Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa.

1:30 P. M. "Reducing the Bearing Cost per Ton of Steel Rolled" by K. T. MacGill, Joseph T. Ryerson & Son, Inc., Chicago, Ill.

2:00 P. M. "Celoron Bearings for Roll Necks" by A. J. Schmitt, Continental Diamond Fibre Company, Newark, Del.

2:30 P. M. "Non-Metallic Roll Neck Bearings" by R. R. Titus, President, Synthane Corporation, Oaks, Pa.

3:00 P. M. Lubrication Question Box:

Cranes, Roll Necks, Filtering Apparatus, Lubrication Devices.

Thursday Evening, June 23, 1932

Formal Reception, Main Ball Room
William Penn Hotel

General Chairman, J. J. Booth, Superintendent, Electrical Department, National Tube Company, Gary, Ind.

IRON AND STEEL EXPOSITION

William Penn Hotel **June 20-23, 1932**

EXPOSITION HOURS

Monday	1:30 to 5:30 P. M.	Evening 7:00 to 10:00 P. M.
Tuesday	1:30 to 5:30 P. M.	Evening 7:00 to 10:00 P. M.
Wednesday	1:30 to 5:30 P. M.	Evening 7:00 to 10:00 P. M.
Thursday	1:30 to 5:30 P. M.	Evening 7:00 to 10:00 P. M.

Wanted—
A Leader

▲

By W. H. WARREN
Vice President, Lukens Steel Company, Coatesville, Pa.

A review of general business conditions show that the steel industry is in as good a shape as any to lead the country out of depression. It took courage to take a stand for better prices in a period of general decline, but the steel business, having plunged to the bottom knows there is no remedy there and that the only way out is up.

W. H. WARREN

But we have only started. If we are not to slip back we must look the facts squarely in the face and the industry must obtain a fair return on its investment. Sales and production departments must work together. Keeping production in line with actual, not imaginary demand, means fair prices. Producing more than the market can absorb means lower prices and less money. Being too proud of shrewdness, and sharp dealing is sure to rebound with disastrous consequences. Someone has sententiously remarked "What does it profit a man if he lose the whole world and become a pauper?"

Conditions today call for a common front, for united action for the general good.

But we are only one industry which is suffering due to nation and world wide inertia. The farther we go the deeper we seem to sink in the mire. Financially, commercially, and industrially, we are fog-bound and we are vaguely conscious that new conditions have arisen. All of the old landmarks have disappeared and we are at a loss to understand why our old principles and remedies do not apply as satisfactorily as they once did. The steel industry is just one cog in the machine, and has to labor or not to labor according to luck and await the resurrection. In addition, every man who works has to pay for every man and woman who is not working. This is not a political question and we have no right to treat it as such. However, no question which touches the prosperity and well-being of a country is political, as such, and no man or men who hold to any doctrine or dogma which stands or may stand in the way of recovery should be allowed to determine the destiny of a country. We are perhaps, transgressing a little in our discussion, but it is difficult to divorce the welfare of industry from the effects of what is known as political action.

We do not believe that this country is devoid of brains of such calibre and clarity which would find a way out of the mire in which we are floundering. We have such men, but they cannot intervene while our present leaders are muddling along, refusing in their selfish dogmatism, to test out different methods of treatment.

If only one man of standing, experience, and ability would call upon the country to end this regime of incompetence and inertia, lifting the problems right out of the political plane, we think that thousands of men engaged in commerce and industry would enthusiastically support his lead.

It does not require mathematical solution to show that the workers produce this Country's wealth and spend it, either in Finance, Commerce, and Industry, but the greatest of these is Industry.

To The Association of Iron and Steel Electrical Engineers

◆

By Gerard Swope, President, General Electric Company, Schenectady, N. Y.

Your celebration this year is not merely the observance of a quarter of a century of existence as an organization, but more essentially is an indication of innumerable obstacles encountered and overcome, of problems presented and solved.

Today, on your twenty-fifth anniversary, we are faced with problems of such magnitude as to suggest radical departures from long-established industrial practices. Before we can clearly point the way to future achievement, we must meet the economic situation which confronts us, and meet it adequately. Unemployment and its attendant difficulties must be cleared away.

GERARD SWOPE

To the members of the Association of Iron and Steel Electrical Engineers, whose talents for twenty-five years have been directed toward the surmounting of industrial difficulties, is offered the opportunity to join their abilities with those of similar industrial groups in an effort to solve our common problem.

Mechanical Power Transmission

By J. F. KELLY,
Managing Director, A. I. & S. E. E.,
Pittsburgh, Pa.

While it has been generally recognized throughout the life of this Association that its affairs have been connected, primarily, with the problems of application of electric power to the steel industry, it is felt that it might be well worth while to consider taking up the entire question of power transmission in the industry as one of the future activities.

Since it probably would not be desirable to take this question up through a separate section, such as the combustion section for handling combustion problems, it is felt that it could be handled quite satisfactorily through the medium of a Power Transmission Committee. This committee would be chosen from those in the membership who are in a position to contribute most to the successful study of this problem.

There are in the Association membership at the present time, many active members who originally were connected with the electrical departments of the various plants but who have since had their duties broadened to the extent that their activities now have more to do with the power transmission problem than the mere electrical application. Therefore, these men would be in a position to do valuable work in connection with the study of the power transmission problems.

An analysis of this question reveals the fact that to properly study the problem will probably require several divisions of the so-called Power Transmission Committee, as there is something over a dozen separate and distinct means of transmitting power, many of which are in common use in the mills today. Without going into details of who could best serve on this committee, it can readily be seen that a member properly qualified to serve on a committee considering the transmission of power by means of a gear train, might have very little to contribute to the question of transmitting power hydraulically or magnetically. Therefore, taking into consideration the various ways which power can be transmitted, considerable care will have to be exercised in choosing the members who are to serve on this committee if the best results are to be obtained.

While it is the plan to make this organization of a permanent nature, it may be well to consider proceeding slowly, at first, with only one branch of this committee functioning at the start.

With a view of exploring the field in an attempt to establish to what extent such a committee could be of service, it has been suggested the first activities of the committee be confined to specifications covering reduction gear drives. The work of this committee should not be confused with the work of any of the standarization committees as no attempt will be made to standardize this class of equipment.

The foregoing is submitted for your consideration as it will come up as an item for general discussion during the June convention, and your comments as to the advisability of proceeding along the lines as outlined above, will be appreciated. It is therefore suggested that you give this matter some thought and come prepared to express your opinion one way or the other on this question.

Electrification of the Iron and Steel Industry

By J. S. TRITLE
Vice President and General Manager, Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

The Association of Iron & Steel Electrical Engineers is to be congratulated upon the completion of twenty-five years of very valuable service to the Iron and Steel Industry. Your progressive policy in the study and discussion of new developments and timely engineering problems accounts for your broader and more prosperous organization of today.

When your Association was formed electrification of the Steel Industry was confined to lighting and motorization of a few small auxiliaries. The Electrical Department was of minor importance in the general operation of the plant. In the twenty-five years of your existence many important developments have occurred, many of which have been of such character as to radically change the process of the manufacture of steel. Today the Electrical Department is not only one of the most advanced in steel plant organization but no projects of any magnitude can be developed without the assistance of the Electrical Department.

An organization such as the Association can be of great value during periods of limited operation such as we are now experiencing. New ideas can be introduced that will effect greater economies of operation and improved product. The development of such ideas is of great importance at the present time.

J. S. TRITLE

With the same enthusiastic effort that has been shown in the past it will be possible to develop opportunities, many of them requiring but a small amount of capital, of introducing new economies that are attractive even with the present day limited production. Nothing more effective in our present day situation could be given than contributions of this nature. We are sure the members of the Association of Iron & Steel Electrical Engineers will contribute their share in any program of this nature.

▲

The Electric Utilities and the Steel Industry

By F. R. PHILLIPS
President, Duquesne Light Co., Pittsburgh, Pa.

At the early meetings of your Association, twenty-five years ago, it is unlikely that there was much discussion of the place of purchased power in your operations. In 1907 the electric utilities were just beginning to explore the possibilities of industrial markets, and a separate power plant for each mill and factory was still the accepted rule.

The situation is quite different now. Central station service has won a large place in industry. Against the stiffest kind of competition—cold, practical, economic analysis—its headway during the past fifteen years has been one of the significant aspects of our times.

Naturally, utility management would like to see every isolated plant in its territory dismantled and its own company's service used wherever electricity is needed. But utility executives realize, although an understandable zeal for new business sometimes clouds the judgment of the best of us, that in the long run central station service must justify itself on economic grounds; and that under certain circumstances such justification does not exist.

High pressure methods may succeed in selling the service, but if it is not satisfactory, measured by whatever yardsticks may be applied, no amount of super-salesmanship will keep it sold. And in the

F. R. PHILLIPS

sale of power to large industrial establishments, where heavy investments in distribution facilities for the exclusive use of the customer may be necessary, it is as important for the utility as for the customer to be assured of the economic soundness of the arrangement.

For reasons more familiar to you than to me, the heat balance in the basic operations of steel making permits the local generation of electric power at costs with which central station service often cannot compete. The margin between the costs of power from these alternative sources is certainly narrower than it once was, but it is entirely possible that the ordinary relations of seller and purchaser may never prevail between the utilities and the larger producing units of the iron and steel industry. Instead, it is not unlikely that the heat and power requirements of such producing units may be integrated and coordinated with those of the utilities through carefully worked out interchange agreements.

What would otherwise be waste heat is not always available for conversion into kilowatt-hours, and in many instances very careful analysis may be necessary to determine the relative economy to an industrial establishment of producing or purchasing its electric energy requirements.

Too often, it seems to me, such analyses give undue weight to relative costs under normal production schedules and tend to ignore a number of factors that cannot well be reduced to dollars and cents upon a statement of comparative operating expenses.

Not least among the old principles to which the present depression has given new significance is the wisdom of the long term point of view. We should not permit conclusions based on normal conditions to obscure the fact that conditions are not always normal.

Many enterprises which have been purchasing utility service rather than producing their own power are now realizing an advantage that probably was not even considered when the choice of power sources was made. With production at low levels due to current business conditions, their power requirements are correspondingly low; and because of the use of utility service, such organizations do not have to bear in these difficult times the burden of fixed charges upon their own power producing facilities.

For the privilege of owning its own power plant, a mill or factory must make a long term committment—possibly twenty to twenty-five years. A contract for utility service on the other hand, is ordinarily an obligation of three to five years' standing at most, and after the initial term has elapsed is usually a year to year arrangement. Moreover, central station service may be had to the extent actually needed, may be kept in step with either increased or decreased power requirements. When industrial operations are curtailed, the demand charges for central station service are invariably much less than the fixed charges on an isolated plant.

It is true, of course, that the average cost per kilowatt-hour of purchased power tends to rise as the user's load and load factor declines. It is also true that some of the more rigorously drawn utility rate schedules base the customer's "demand charge" for any given month not on the actual power demand created during the month but on the maximum demand established during the preceding year. In a long continued economic depression such as we are now experiencing, however, even this form of rate affords substantial relief in the over-all cost of power, whereas the only possible reduction in the total power cost of an enterprise owning its own power plant is found in rather nominal savings in fuel, labor and maintenance expense. It is doubtless a fact that a great many isolated power plants which can show an economic advantage over purchased power when production is normal cannot show such an advantage over a number of years during some of which production is markedly below normal.

What I am suggesting, of course, is that in any consideration of alternative power sources we adopt a long term point of view. This depression may be the last—but it probably won't be. And where a careful analysis of a given situation discloses only a rather narrow margin of costs in favor of isolated plant operation within the normal production range, it may well be that a reasonable recognition of relative power costs during the inevitable periods of reduced activity may swing the balance in favor of central station service.

Because it appears impossible to predict with any assurance of accuracy the arrival, duration and degree of our alternate periods of prosperity and depression, it would be very difficult to obtain any quantitative measure of the possible advantage of utility service during periods such as the present; nevertheless, the great preponderance of fixed charges in the total cost of electric energy is such that the probable necessity of maintaining a power plant in partial idleness during periods of depression is certain to be reflected in the long term average cost of that energy.

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Chairman, Cleveland Section,

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LINN O. MORROW,
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W. W. SPANAGEL,
Secretary, Cleveland Section,

M. J. CONWAY
Chairman, Instruments and Control Section, Combustion Engineering Division.

G. R. CARROLL
Standardization Committee.

F. L. GRAY
Secretary, Lubrication Session, Thursday, June 23, 1932.

J. H. VAN CAMPEN
Chairman, Invitation and Ticket Committee, Convention and Exposition.

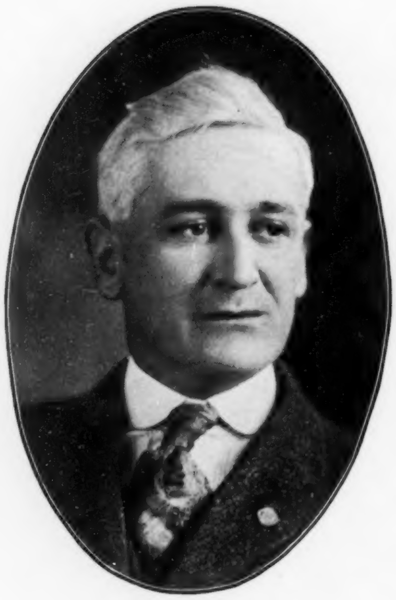
W. N. FLANAGAN
Gas Utilization Section, Combustion Engineering Division.

F. D. EGAN
Chairman, General Registration Committee.

F. W. CRAMER
Chairman, Ladle Crane Committee.

F. O. SCHNURE
Chairman, Electrical Engineering Session, Tuesday, June 21, 1932.

H. F. RAAB,
Secretary, Pittsburgh Section



Prominent Steel Mill Engineers, who will be active in the various Technical Sessions of the A. I. & S. E. E. Convention and Iron and Steel Exposition at Pittsburgh, June 20-23, 1932.

J. D. DONOVAN,
General Convention Chairman.

W. E. MILLER,
Chairman, Iron and Steel Exposition Committee.

J. F. HUNTER,
Chairman, Safety Engineering Session, Monday, June 20th.

G. T. HOLLETT,
General Chairman, Combustion Engineering Session, Wednesday, June 22nd.

C. C. PECU,
General Chairman, Lubrication Engineering Session, Thursday, June 23rd.

GROVER A. HUGHES,
Chairman, A. I. & S. E. E. Welding Committee.

A. C. Cummins Made Assistant General Superintendent Duquesne Works, Carnegie Steel Co.

A. C. CUMMINS

Alden C. Cummins, who has been Director of Power Distribution and Superintendent of the Electrical Department of the Duquesne Works and one of the prominent members of the Association of Iron & Steel Electrical Engineers, has been made Assistant General Superintendent of that plant of the Carnegie Steel Company. Mr. Cummins succeeds David C. Burroughs, who resigned January 31, 1932.

"Al", as he is generally known, is a native Pittsburgher. He attended the public schools in Pittsburgh and also Shadyside Academy. He entered Lehigh University in 1906 and graduated in 1910, receiving his degree in electrical engineering with additional credits in mechanical engineering and mathematics. During the summer vacations of his college course he served as an electrician with the South Penn Oil Company at Folsom, West Virginia.

In July, 1910 he accepted a position with the Westinghouse Electric & Manufacturing Company and in January, 1911 he entered the service of the Carnegie Steel Company as a draftsman. Later, he became Assistant Superintendent of the Electrical Department and then Superintendent of the Ferro Alloy Furnace and Electrode Factory. He succeeded Carl Cederland as Superintendent of the Electrical Department of the Duquesne Steel Works. Later, he was given control of all inter-works transmission lines between Carnegie plants in the Monongahela Valley with the title of Director of Power Distribution.

Mr. Cummins has been a member of the Association of Iron & Steel Electrical Engineers for fourteen years. He has served on practically all of the national committees of the Association. He was Chairman of the Pittsburgh District, later becoming a Director and in 1925 was elected President of this Organization. He was Chairman of the Standardization Committee in 1926, 1927 and 1928 and during that time specifications were prepared and adopted for the standardized commutating pole mill type motor. He is at the present time the Chairman of the National Meetings and Papers Committee of the Association.

Mr. Cummins has contributed to the transactions of the Association of Iron & Steel Electrical Engineers many valuable papers and discussions, among the outstanding ones being:

Developments in Electrical Maintenance Shop Practice.
The Standardized Commutating Pole Mill Type Motors.
Experiences in the Steel Industry with the Grounded Neutral.
Standardization Committee's Final Outline Dimensions for Association of Iron & Steel Electrical Engineers Standardized Mill Type Motor.
The Engineering Industry.

Mr. Cummins is a member of the American Institute of Electrical Engineers and is serving on a number of their national committees. He is also a member of the Engineers Society of Western Pennsylvania, American Electro-Chemical Society and a member of the Board of Education of the City of Duquesne.

The President and Board of Directors and the Membership of the Association of Iron & Steel Electrical Engineers wish to extend at this time to Mr. Cummins their sincerest congratulations on his advancement to Assistant General Superintendent of the Duquesne Works of the Carnegie Steel Company at Duquesne, Pa.

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Annual Review Electrical Developments 1931–1932

By W. H. BURR
Chairman, A. I. & S. E. E. Electrical Developments Committee and Electrical Supt., Lukens Steel Company, Coatesville, Pa.

To be presented before A. I. & S. E. E. Annual Convention in Pittsburgh, June 20-23, 1932.

From an informal meeting of twenty-eight Electrical and Mechanical Engineers in 1907 to an Association of over twenty-one hundred members in twenty-five years, with sections in five of the principal cities of the United States, is a development which should be an inspiration to every member of this Association.

To the twenty-eight men, many of whom are still very active in the Association, we more recent members are greatly indebted. To the manufacturers of electrical equipment who have co-operated through all these years in all the activities of the Association and to the steel companies who have sponsored our activities, we are indebted in perhaps no less degree.

W. H. BURR

Yearly reports of developments are given, but not until we are confronted with the record of a period of years do we realize to the fullest extent what wonderful strides have been made in the Electrical Industry and what a vital part the Association has played in the highly developed art of steel making.

The Association had reached a stage in 1916 where it was necessary to adopt new policies in order to broaden the scope of its activities. In 1931, the steel industry underwent many changes, and after this depression is over the benefits derived from the lessons learned will no doubt result in greater achievements than ever before in this industry.

The year 1931, lean as it has been from a financial standpoint, can boast of a very productive year in electrical developments. With a reduced operating program, consideration has been given by the Steel Industry to many problems which were heretofore overlooked in the race for mass production.

Labor has contributed very materially and executives have labored as never before with problems which have confronted the steel industry during the past year. Many of the problems are still unsolved but the lessons learned will have a very important bearing on the future of the industry.

In the solution of its operating problems, whether they be heating, rolling, annealing, welding or transportation, the electrical industry is destined to play a leading part.

STEAM TURBINES

In 1899, there were built the first American turbo-alternators for a speed of 3600 r.p.m.; and for years thereafter the ideas of speed were considered excessive by those who felt that 750 r.p.m. was the proper figure for large units. Less than ten years ago, the largest turbo-generators ran at 1200 r.p.m., moderate size ones at 1800 r.p.m., and only the small ones of 7500 kv-a. or less at 3600 r.p.m. Now 1800 r.p.m., has superseded 1200 r.p.m. for even the biggest units, and is itself being pushed from below by the still higher speed. Two years ago, a 12,500 kv-a., 3600 r.p.m. machine was notable; last year one of 18,750 kv-a. was reported; and even today's 22,500 kv-a. unit is not the limit, for designers feel able to undertake generators of this speed up to more than 30,000 kv-a. with internal fans, or double that figure with such aids as external fans and aluminum field coils. The substantial advantage is in space, weight, and efficiency.

In the past decade turbine-generators have grown bigger astonishingly, each year's biggest is soon eclipsed by a bigger yet, until there are now units around 200,000 kv-a. Even this is not technically a limit —designers feel that 1800 r.p.m. generators of 250,000 kv-a. are possible in a single unit; and three-cylinder turbines of 360,000 kw. have been proposed. Pressures have gone up too, with 1200-1500 lb. in service, and 3200 lb. considered. Larger sizes and higher pressures yield advantages in thermal efficiency and disadvantages in complication and cost.

Two 2000 kw. turbo-generators for the Dupont Company represent a curiously interesting return of high frequency power. It was as high frequency (133 cycles) that the alternating current first made electric service generally available forty-six years ago.

The 180 cycle energy of the new generators will be used, not for general service, but for extremely highspeed spinning buckets which make artificial silk—buckets driven by two-pole, 10,000 r.p.m. induction motors. Frequencies down to 120 cycles are available by speed variation of the turbine.

MERCURY-VAPOR TURBINES

Two new mercury-vapor power-plant equipments were being built and each will have a generating capacity of about double that of the apparatus previously built.

The mercury parts of both the new installations will be alike. The main difference in the two plants is that at Kearny the mercury turbine-generator and condensers will be installed above the furnace setting, thus providing gravity feed for the mercury boiler; while at Schenectady the turbine unit will be installed alongside the furnace and the mercury returned to the boiler by a mercury feed pump. This arrangement with feed pumps makes possible an outdoor power station, which reduces building and structural costs and facilitates future extensions.

The mercury turbine of each of the new equipments is a double-flow unit of 20,000 kw. capacity, operating at 900 r.p.m. About 240,000 lbs. of steam per hour will be generated by condensing the exhaust mercury vapor and this steam will be superheated to 750 deg. F. total temperature in a convection superheater installed above the mercury boiler.

TURBO-GENERATORS

The 160,000 kw. unit installed in the Hudson Avenue Station of the Brooklyn Edison Company is a tandem-compound turbine driving a single generator at 1800 r.p.m. The steam at 400 lb. pressure and 730 deg. F. total temperature enters the high-pressure turbine and exhausts from it into a double-flow low-pressure turbine at a variable pressure determined by the load on the unit. The 160,000 kw., 200,000 kv-a. generator is more than 50 per cent larger in capacity than any previous 1800 r.p.m. generator.

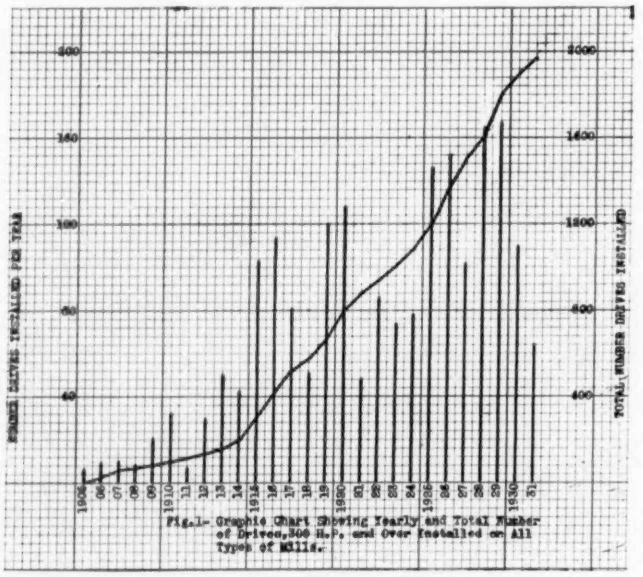

Fig.1- Graphic Chart Showing Yearly and Total Number of Drives,300 H.P. and Over Installed on All Types of Mills.

SYNCHRONOUS CONDENSERS

The idea of improving the power-factor and hence the usefulness of a system is not new, and synchronous motors have been used for that purpose for years in various industries. Such machines for industrial use have been relatively small. It is only since engineers have approached the problem of giving long transmission lines maximum stability and transmitting capacity that synchronous condensers have been appearing in astonishing sizes.

A 75,000 kv-a., 514 r.p.m. unit being built for the Commonwealth Edison Company is larger by a half than any other. It is to be connected through transformers to the 220 kv. line which transmits 150,000 kw. into Chicago (Crawford Avenue Station) from the Powerton Station, 150 miles away. The machine is 40 ft. long, 16 ft. in diameter and weighs over 300 tons. The main machine is fitted with a direct-connected, 3750-hp. induction motor for starting and synchronizing. This motor is also used to drive the condenser as a generator to supply voltage for testing the transmission line.

MERCURY RECTIFIERS

Despite the various improvements that have been made, the mercury rectifier has, until quite recently, lacked the reliability demanded by commercial service in America.

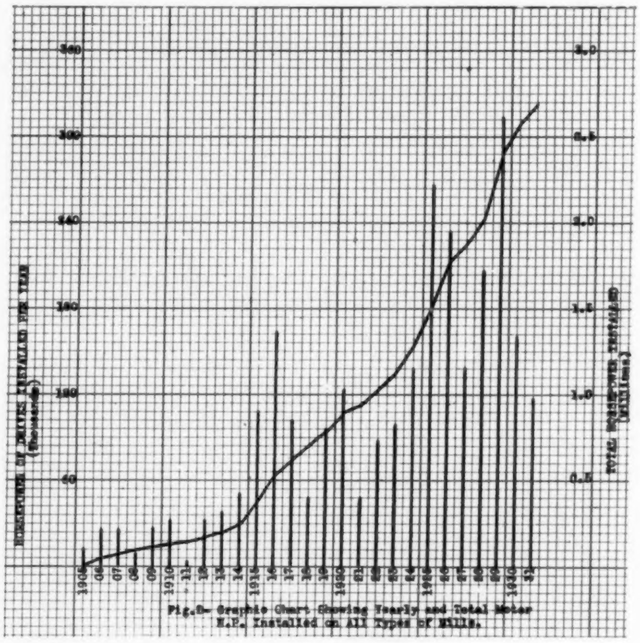

Fig.2- Graphic Chart Showing Yearly and Total Motor H.P. Installed on All Types of Mills.

Fundamental investigation has brought about such a change and rectifiers have been built, tested, and put in service, which are now as reliable as a synchronous converter.

TRANSFORMERS

Engineers have become accustomed in the past few years to very large transformers. Nevertheless, several units built this year are big enough to claim renewed attention.

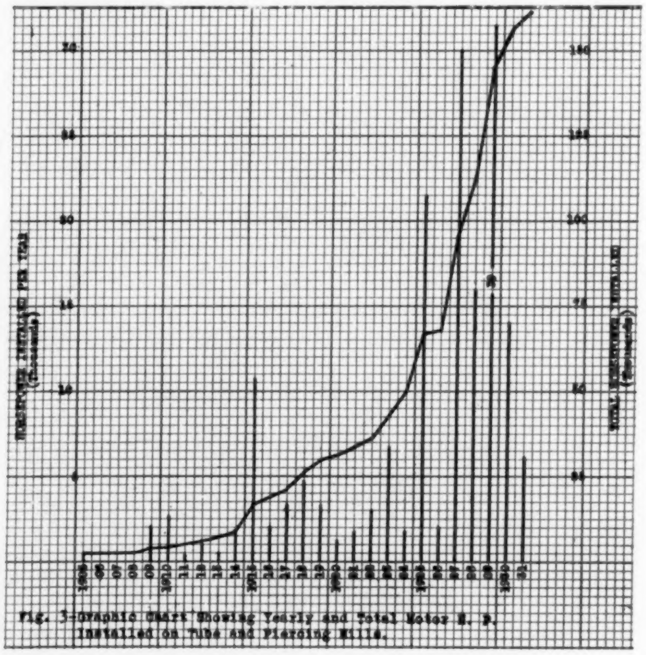

Fig. 3-Graphic Chart Showing Yearly and Total Motor H. P. Installed on Tube and Piercing Mills.

The largest three-phase transformer ever built in America, 70,000 kv-a., 60 cycles, went to the United Electric Light and Power Company of New York

City. Motor-driven blowers aid the ventilation of its rows of radiators, which are mounted on headers, connected to the tank with large pipes. The largest piece, weighing 110 tons, could not pass through railroad clearances in New York and so had to be unloaded in Jersey City, and taken to Hell Gate by water. The transformer weighs 213 tons and covers a floor space of 20 by 37 ft. Its voltages are 132—14 kv. Since both windings will be connected to underground cables, it lacks the tall bushings usual on transformers, their place being taken by potheads. But even thus truncated, the height is 28 ft.

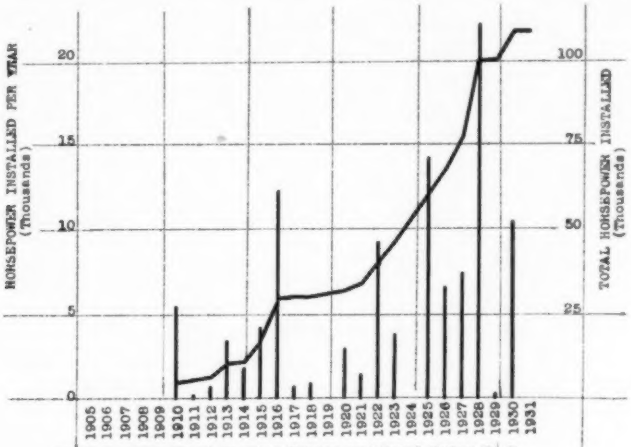

Fig. 4—Graphic Chart Showing Yearly and Total Motor H. P. Installed on Rod Mills.

Surge-Proof, 42,000 kv-a., single-phase units—four of them—were built for the Consolidated Gas, Electric Light & Power Company, of Baltimore. These transformers are among the largest 60-cycle, single-phase units yet built, and in overall dimensions they are but little smaller than the largest at any frequency. Each occupies a floor space of 21 by 22 ft., is 35 ft. high overall, and weighs 190 tons complete. They will step down 220,000 volt power from the Safe Harbor transmission line to 33,000 for use in the Baltimore district.

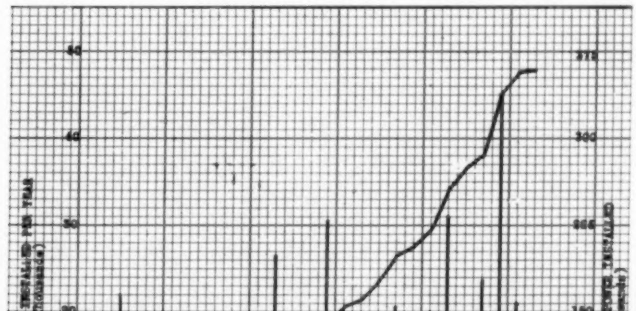

Fig. 5—Graphic Chart Showing Yearly and Total Motor H. P. Installed on Bar and Billet Mills.

The ability to withstand artificial surges simulating severe lightning is, however, the most interesting distinction of the newest transformer designs.

As the general reliability of power systems has increased, outages due to lightning have become more conspicuous and have provoked much study. A nearby stroke produces stresses on transformer insulation, not only greater, but different in kind from ordinary forms of electricity; piling up voltage at an astonishing rate. In one case a natural lightning, probably the most severe yet recorded, this surge rose at the rate of 4,000,000 volts per microsecond (one-millionth of a second).

As a result of several years' lightning studies, however, a new method of design and insulation has been developed which appears proof against any such surges so far observed. A three-million-volt surge generator, or "lightning generator", was built and with it these transformers received the most severe tests a commercial transformer has ever been given.

At 1,100,000 volts, a standard 64 in. protective gap flashed over—but the transformer did not. At 1,300,000 volts, a string of 14 in. suspension insulators flashed over—but not the transformer. At 1,500,000 volts, the transformer bushing flashed over—but the transformer windings and insulation withstood it.

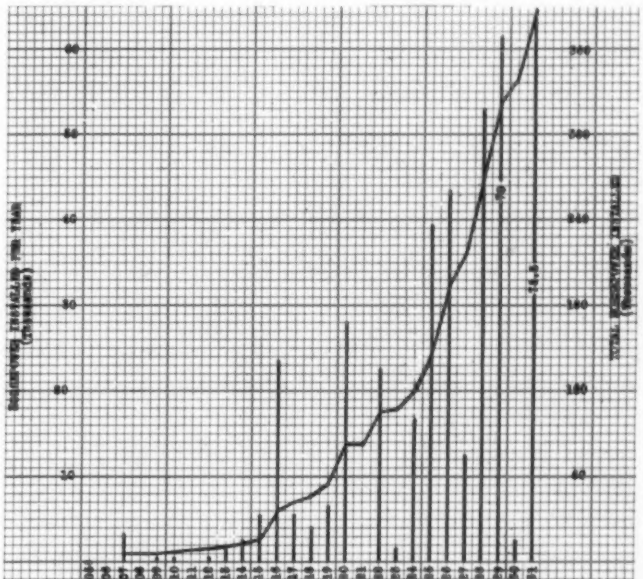

Fig. 6—Graphic Chart Showing Yearly and Total Motor H. P. Installed on Strip Mills.

For a final and most severe test, a surge of 2,520,000 volts was thrown on the transformer, rising at the rate of 11,000,000 volts per micro-second, a rate nearly three times that of the most severe observed lightning. The transformer survived uninjured and afterwards passed the regular commercial tests.

One line of distribution type transformers has been developed which has circular coils, down to the smallest size, and the coils are all of the open type construction. This means that the individual layers of high tension and low tension windings are separated from one to another not only by solid insulation, but also by vertical ducts.

This construction permits treatment of coil in varnish of the same grade as used on large power transformers. The open ducts permit air to be circulated through the coil and to oxidize the varnish, thus giving the coil the same desirable mechanical strength and cooling qualities as on large power transformers.

The small temperature gradient between copper and oil permit heavy overloads without injury.

MAIN MILL DRIVES

In looking over the records of main roll drives sold during the year 1931, we find that the total was only 63, or about half as many main roll drives as were sold during the previous year. The total horsepower represented by these motors was 119,325, somewhat more than one-half of last year's total. Not since 1923 has this total horsepower dropped to such a low level. This brings the total horsepower installed on Main Roll Drives up to approximately 2,700,000.

The increase in the horsepower rating of the individual motors was also notable, there being ten 3500 hp., one 4000, one 4500, two 5000, and two 7000 units in the list of those purchased during 1931.

The statistics show very plainly a trend toward higher voltages on the a-c. drives for there were four 2200 volt, ten 6600 volt, two 10,000 volt and one 11,000 volt a-c. motors purchased for steel mill drives. The proportion of geared units to direct drives greatly exceeded that of any previous year, there being 60 geared and 3 direct connected.

The number of motors purchased for Strip and Hoop mills (29) greatly exceeded that for any other type mill and in fact represented nearly half the total number purchased. Cold Roll mills were second with 12 installations and Merchant mills were next with a total of 11 motors.

Statistics show that one-fifth of the total horsepower of motors purchased during the past year went to the Amtorg Co. and were sent to U.S.S.R. The very noticeable decrease in sales of motor drives during the year, serious as it has been, is not in my opinion to be compared with the seriousness of this situation.

MAIN MILL DRIVES—D.C. MOTORS

Nearly a year of operation of a 10,000 hp. "Twin Motor" drive on the main rolls of the 44" slabbing mill, and for a longer period of a similar 10,000 hp. drive on the 54" blooming mill at the same plant, has demonstrated the advantages of this method of drive over the usual single motor and pinion stand drive. Pinion losses, repairs and maintenance are eliminated; the motor inertia can be reduced resulting in reduced motor and generator heating, more torque available for rolling work, and faster acceleration; roll chatter, backlash and spindle vibration are practically eliminated; and higher rolling speeds can be maintained and the mill output increased. This method has proved particularly advantageous for these large reversing mills requiring unusually high torques, but analyses recently made indicate that it is also entirely suitable for smaller reversing mills and for some types of continuously running mills.

A large producer of sheet and tinplate has installed a single stand cold roll mill and tension reel to produce tinplate in length coils. The mill stand is driven by a 675 hp., 533/800 r.p.m. motor and the reel by a 500 hp., 300/900 r.p.m. motor, both operated on variable voltage from an existing spare motor generator set. The supply reel is equipped with a back tension generator feeding a motor on the 675 hp. mill motor shaft, thus maintaining tension on the strip entering the mill to eliminate buckles and reduce the tendency to cobble.

A three stand tandem cold roll mill for rolling high grade alloy steel was placed in operation the first part of the year. Each stand is driven by a 300 hp., 400/1200 r.p.m., direct-current motor which receives power from two motor generator sets. Variable voltage starting and speed adjustment of the whole mill is used. Tension between stands is regulated and maintained by using series exciters which excite a combination of differential and cumulative variable potential fields on the motors. A special regulator is used to maintain tension between the last stand and the reel.

A three-high universal plate mill, driven by a 4000 hp., 40 deg., 80/160 r.p.m., adjustable speed motor has been installed in the Southern district. While the motor is only reversed in case of emergency, both motor and control are of standard reversing type construction. This control permits entering a slab in the mill at comparatively slow speed and then accelerating and finishing the pass at a higher speed, thus increasing the possible production from the mill. The power for the motor is supplied by a motor-generator set consisting of two 1750 kw., 40 deg., 514 r.p.m., 800 volt generators, driven by a 4700 kv-a., 6600 volt synchronous motor.

In the Cleveland district, there is being installed a 72 in. wide continuous hot strip mill. The drive for this mill consists of one 800 hp., one 1200 hp., one 1600 hp., one 2000 hp., and one 3000 hp. induction motor, one 2500 hp. synchronous motor, three 3500 hp. and one 2500 hp. direct-current motors. Direct-current power is supplied by two 4000 kw. motor-generators.

The same company is also installing a two stand, 72" tandem cold strip mill believed to be the widest cold strip mill so far built. Each stand of this strip mill will be driven by a 1500 hp., 200/400 r.p.m., 600 volt, d-c. motor. These are the largest motors ever used in cold strip mill drives. The tension reel will be driven by a 400 hp., 300/1050 r.p.m. motor. This large capacity in the reel motor is required simply to place the strip under tension between the last stand in the mill and the reel in order that it may be coiled without wrinkles.

The control equipment for this mill is so arranged that the reel motor at all times automatically maintains the strip under tension, regardless of whether the mill is at standstill or is running. The whole mill may be automatically slowed down to a predetermined slow speed for threading the strip through the mill, and then speeded up to the running speed. This change from threading to running speed is made by generator voltage adjustment.

Because Canadian railroads are demanding longer and heavier rails, an existing steam-driven mill has been rebuilt and electrified, and can now turn out rails of the heaviest section (130 lb. per yard) fast enough to lay 195 miles of track monthly.

The new mill, consisting of a 30 in., three-high, pinion stand and three stands of rolls, is driven by a 5000 hp., 50-120 r.p.m. reversing motor, the largest direct-current machine ever built in Canada. It is fed with current from a set consisting of a 4250 hp. induction motor, two 2000 kw., 700 volt generators, and a 75,000 lb. steel plate flywheel.

An interesting drive was built, during the past year, for a "push bench" to produce seamless steel tubes. In the production of tubes by this method, a hot billet of steel is first placed in a hydraulic press and formed into the shape of a thick-walled thimble. This thimble is then placed on the end of a mandrel and is forced through a succession of stationary ring dies which gradually decrease its wall thickness and elongate it, forming a closed-end tube. The mandrel is then removed and the closed-end sawed off, after which the tube is finished in the usual manner.

The mandrel is forced forward through the dies by means of a rack and pinion driven by a 1500 hp., 150/300 r.p.m., 600 volt, direct-current, reversing motor. Power is supplied to this motor from a 1250 kw. synchronous motor-generator set. The novel features of the equipment are in the control. This is arranged so that, upon indication from the operator, the motor will start and accelerate the ram, force it through the series of dies, stop, reverse, and return to the initial position at a high rate of speed. Means are provided for adjusting the speed of the strokes over a wide range. Extremely fast acceleration and deceleration is necessary to meet the desired cycle.

Following the general trend, one large producer of steel strip in the Cleveland district completely revamped a large number of cold strip mills. Two four stand tandem mills were installed together with electrical equipment. These mills roll steel with tension between stands and are provided with special variable voltage reel drives for coiling the strip under tension as it leaves the last stand. Forty-five single stand mills were rebuilt and changed from a group drive to single motor drives. Twenty-four bell type electric annealing furnaces were installed. Estimates show that the use of these furnaces will result in a very appreciable economy as well as improved working conditions and greater cleanliness which is an important factor in cold strip mills.

MAIN MILL DRIVES—A.C. INDUCTION MOTORS

A 76 in. continuous hot strip mill, capable of rolling wider strip than any other similar mill so far installed, is being put into operation in the Chicago district. The first stand of this mill is driven by a 1000 hp., 735 r.p.m., 6600 volt, squirrel cage induction motor. Simplicity of motor and control equipment dictated the use of the squirrel cage motor in this instance. The motor is started at reduced voltage obtained by connecting a reactor in the neutral of the motor windings. This is a very unusual application of a squirrel cage motor.

Two pole, sixty cycle, induction motors in the larger sizes, 1000 hp. and above, have been greatly improved and perfected. An outstanding example is the installation of three 1250 hp., 2200 volt, 3 phase, 60 cycle, 3560 r.p.m., squirrel cage motors arranged for starting across-the-line. The kv-a. inrush under locked rotor conditions is only 565 per cent. So far as we are aware, these are the largest two pole motors that have to date been built for across-the-line starting. These machines drive high pressure boiler feed pumps.

A number of two pole slip ring induction motors in 250 and 300 hp. sizes have been developed. These machines are 2300 volt, 3 phase, 60 cycle, 3515 r.p.m. and are arranged for continuous operation over a range of from 2500 r.p.m. to full speed, the hp. load dropping off to 60% at the low speed. The critical speed is necessarily held well above the maximum operating speed. These motors are of the totally enclosed, self-ventilated type. Air is brought to the machines through an intake duct and discharged into a central discharge duct. The fans on the ends of the rotor supply the necessary pressure for the whole ventilating system throughout the speed range. A unique feature is the enclosed collector rings on the end of the front pedestal. The collector housing is supplied with air under pressure from the motor inner end bell and this air, after passing around the rings and brushes, is discharged into the motor discharge duct. The machines are supplied with ring oiled bearings.

It is also believed that this is the first instance of motors of this size and speed being arranged for continuous operation at so great a speed reduction, although there are such machines arranged for starting duty only.

These motors were supplied to drive high pressure gasoline pumps for pipe line service.

Considerable progress has been made in determining improved distributions of squirrel cage windings and more advantageous arrangements of segmental and solid end rings. Motor efficiencies have been materially improved during the past year.

MAIN MILL DRIVES—SYNCHRONOUS MOTORS

Part-winding starting for synchronous motors was increasingly used last year. Two motors for driving pulp grinders are wound with four circuits in the stator which are connected successively to the line to provide starting increments of 375 kv-a. which reduces to a minimum the line disturbance on the power system.

An instantaneous unloader for ammonia compressors has been developed for use with synchronous motor drive to provide automatic by-passing of the compressor. This device consists of a valve operating by ammonia gas compressor differential and interlocked through a pilot valve with the field discharge circuit of the synchronous motor. This unloader permits use of synchronous motor drives for automatic start and stop installations, and permits the motor to carry through voltage dips.

By means of an automatic unloader designed for a triplex reciprocating pump for high-pressure scale removing, a synchronous motor drive was arranged suitable for automatic start and stop operation.

ELECTRIC FURNACES

Unrealized by most people, industrial electric heat has been almost doubling itself each year, and has reached such size that the power of Niagara would not supply all its units turned on at once.

Electric furnaces can produce higher grade steel than any other; but in the days when common steel was good enough, the electric furnace was considered merely an expensive toy. Now, when scientifically concocted alloy and formula steels are required in thousands of tons, electric steel furnaces have become available for producing important tonnage.

The Ford Motor Company put in service this year three electric arc furnaces of the Heroult pattern, each of which can convert 15 tons of molten metal into high-grade steel every hour and a half.

A much larger arc furnace (Heroult type) was sent to Japan. Its capacity, 40 tons, is sufficient to pour from a single batch, the largest piece of a 16 in. gun. The furnace is the largest in Japan—there are few larger anywhere. The transformer which is its power supply is one of the largest of its sort—12,000 kv-a. The furnace is tilted by a 45 hp. series motor to pour out its molten metal, and the three electrodes which feed the arc are each moved forward by a 7.5 hp. shunt motor.

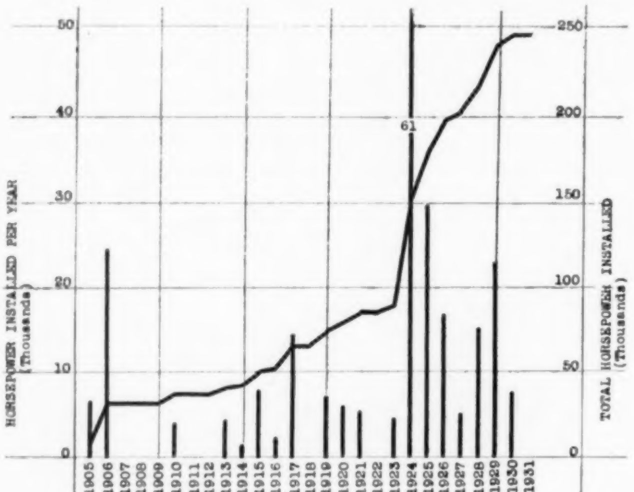

Fig. 7—Graphic Chart Showing Yearly and Total Motor H. P. Installed on Structural Mills.

A new type of furnace, applicable especially to annealing steel sheets, has been developed in which a load of material can be cooled in the furnace in a comparatively short time.

With previous types, the cooling rate of a loaded furnace is so slow as to make the output very small, and the use of furnace cooling impracticable. The new furnace is lined with light weight refractory brick so that the heat storage in the furnace walls is comparatively small, and employs a circulating system for the furnace atmosphere, with an external cooler by means of which the heat may be extracted from the furnace and charged at a comparatively high rate.

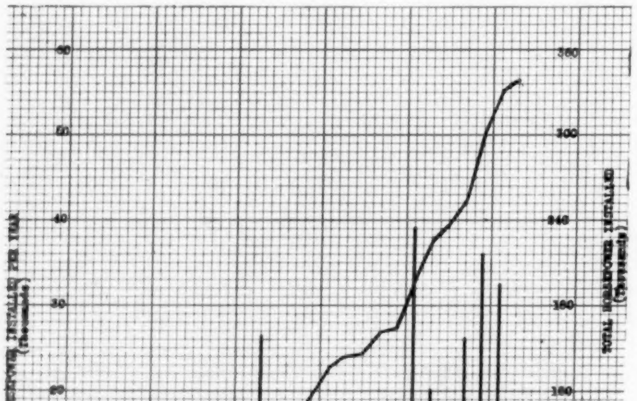

Fig. 8—Graphic Chart Showing Yearly and Total Motor H. P. Installed on Merchant Mills.

The cooling system consists of a finned tubular type surface cooler, through which water flows, the necessary duct and damper system, and a motor driven blower which circulates the gas from the furnace through the cooler and back to the furnace. The cooling rate may be controlled by adjusting the speed of the blower, or the valves in the duct system. The use of hydrogen as an atmosphere increases the effectiveness of the system due to the high specific heat of this gas. A load can be cooled in this furnace in about one-half the time that would be required if the load were set in the open room under a steel plate hood. The surface condition of the load is also improved because it is not necessary to expose it to the air while at high temperature, as would be necessary if it were transferred to a hood. The circulation of the atmosphere insures a uniform cooling rate.

This type of furnace provides means which have not heretofore been available, for cooling masses of material in a non-oxidizing atmosphere at controlled rates.

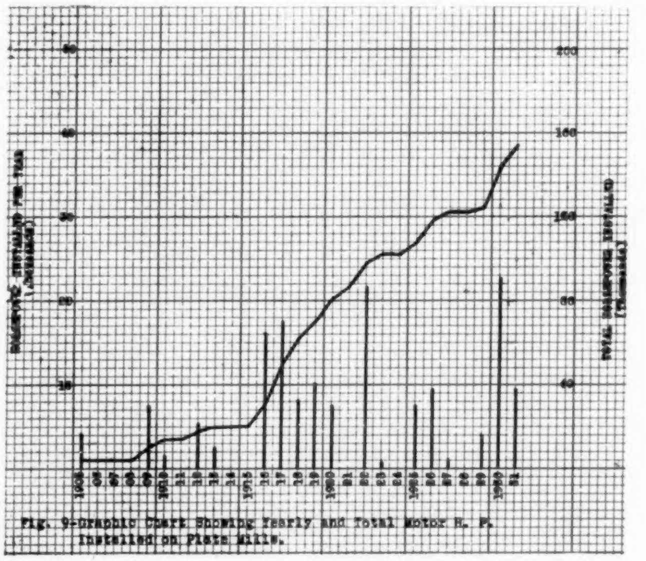

Fig. 9—Graphic Chart Showing Yearly and Total Motor H. P. Installed on Plate Mills.

D.C. AUXILIARY MOTORS

One of the leaders in the development of auxiliary motors has perfected equipment for the rolling of strip steel with a constant tension although the diameter of the roll is changing. Constant current in a d-c. motor armature gives constant torque providing the flux is unchanged. The speed of a d-c. motor varies inversely with the flux. The torque of a d-c. motor varies directly with the flux. To maintain constant tension on a building-up reel, the torque supplied by the motor should be directly proportional to the diameter on which the material is being wound. With the necessary speed change due to the rolled build-up being accomplished by strengthening the flux, constant current in the armature will give the required torque from the motor to maintain constant tension.

This device eliminates the use of a relay for the maintenance of constant current in a d-c. motor armature driving the reel. This is accomplished by placing in series with the driving motor the armature of a control motor, the movement of which adjusts a rheostat in the field of the driving motor to maintain the condition of constant current through the driving motor. There is no movement of armature of the control motor excepting that necessary to properly adjust the rheostat to give the required balance. Several such equipments are in successful operation maintaining constant tension with entire freedom from shut-downs.

A.C. AUXILIARY MOTORS

The outstanding development of the past year has been the application of special alternating current squirrel-cage motors where their use was previously thought impossible. Probably the most interesting illustration is a drive on what is known as roller and catcher tables used for automatic feeding of sheets through sheet mills. These motors are specially designed with low-inertia rotors and provided with Class B insulation. The motors are fully-enclosed. It is essential to have liberal radiating surfaces sufficient to dissipate the losses incident to the oft-repeated reversals. Motors are in use which can successfully be reversed at a rate as high as forty times a minute.

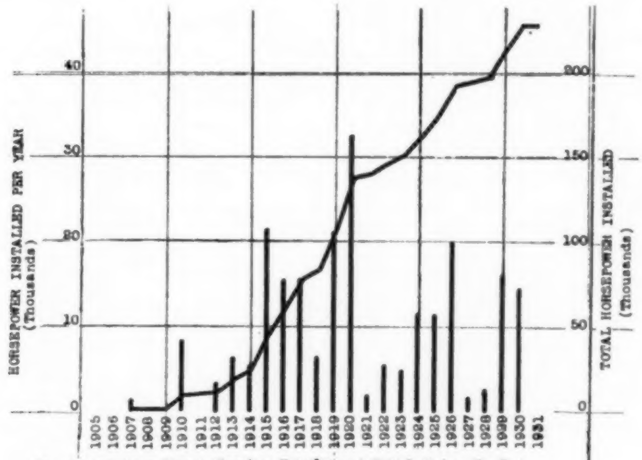

Fig. 10-Graphic Chart Showing Yearly and Total Motor H. P. Installed on Sheet and Tin Plate Mills.

Specially designed, low-inertia motors with from 8 to 10% slip have been applied to many presses and shears where d-c. motors were formerly used. Such alternating current squirrel-cage motors may replace larger motors of conventional design if particular attention is given to the operating characteristics of the motor furnished.

Individual drive of the rollers in a roller table by means of alternating current, squirrel-cage motors has been largely extended during the past year. Of particular interest is a large table with nearly a hundred rollers where quick and accurate stopping is secured by means of direct current dynamic braking. A three-unit, direct current driven motor-generator set supplies the alternating current of varying frequency as well as the direct current used in dynamic braking. The table and control circuits are so sectionalized that the different rollers may be started and stopped in groups.

During the year, there has been considerable use of frequency converters for securing a low frequency on which squirrel-cage a-c. motors may be operated at a reduced speed. The frequency converter is essentially a rotating transformer, the frequency secured depending upon the speed of rotation. The use of a dual frequency offers considerable advantages in special applications.

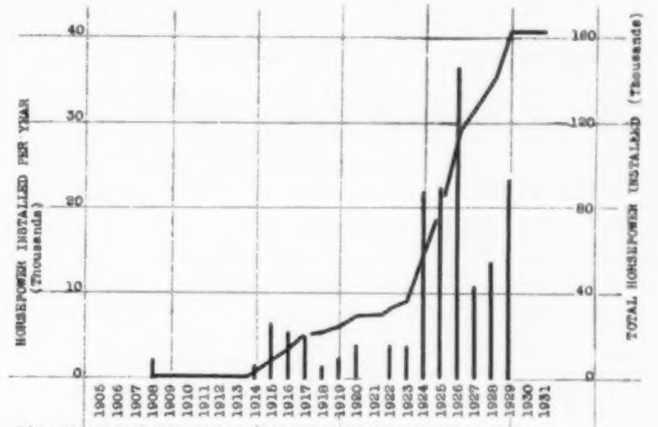

Fig. 11-Graphic Chart Showing Yearly and Total Motor H. P. Installed on Sheet Bar and Skelp Mills.

A one horsepower, direct-current, shunt motor is now operated from the 25 cycle, alternating-current supply, under perfect speed control from standstill to full speed without any rheostatic losses whatever, by the use of four small thermionic tubes and their adjuncts. It is controlled by a foot pedal, which pushes an iron core into and out of solenoid, and thus changes the power-factor of the grids in two of the tubes.

The tubes, two one-ampere rectifiers supplying the field coils and two six-ampere grid glows for the armature, serve to rectify the alternating-current waves; and the tubes, under the action of the grid, also chip off greater or less amounts of the wave, giving a variable effective direct-current voltage from a constant alternating-current voltage, without the usual conversion losses.

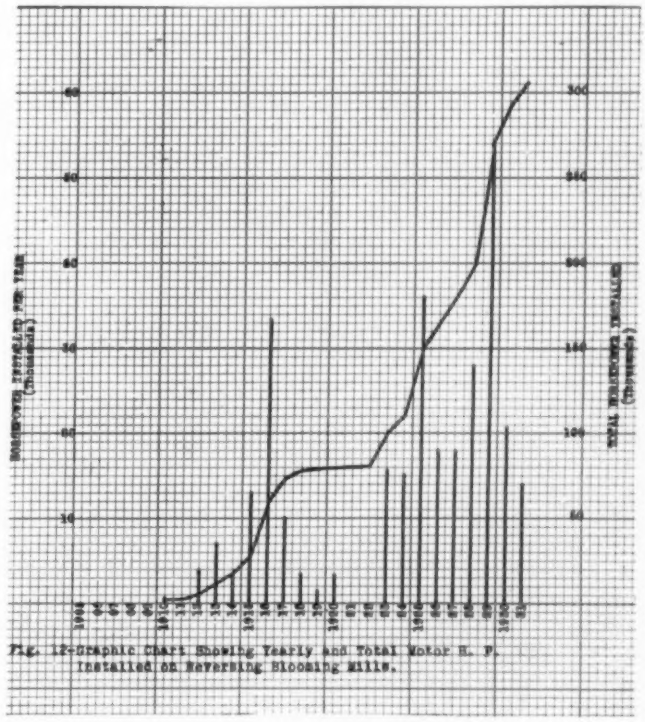

Fig. 12-Graphic Chart Showing Yearly and Total Motor H. P. Installed on Reversing Blooming Mills.

No reason seems evident why similar variable-voltage, direct-current service from alternating-current power should not be applied to industry generally as suitable tubes are available up to very considerable sizes.

It is a well known fact that higher speeds allow smaller size for a given power, but is rather startling, nevertheless, to see a two-hp. motor that can be carried in one hand! Such a motor has been developed, only 4.5 in. in diameter and 2.5 in. long (core length), runs at 18,000 r.p.m., on a 300 cycle supply, and will drive wood-working machinery.

Twenty-five hp. motors so light a man could carry them, 8 in. in diameter, less than 14 in. long (without bearings), were built to drive the propellers of a model airplane in a wind-tunnel test.

Another step ahead in roll out drives was made when an installation of 120 special motor reducer units was furnished for the hot bed rollers of the latest strip mill to be put in operation. These individual motor reducer drives are installed under the apron of the cooling bed and coupled directly to the rolls which handle the hot strip at a maximum speed of 1380 ft. per minute on rollers of 16 in. diameter.

Under these conditions the motors, which are squirrel-cage, ball bearing type, with integrally mounted reducer, are fully enclosed, forced ventilated and Class B insulated. These motors are capable of 9.5 hp. maximum and are energized by a motor generator set designed to give a variation of 450 to 1800 r.p.m. at the motor when varying from 15 to 60 cycles and 110 to 440 volt, 3 phase operation, there being a ratio of 5:1 on the reducers. This gives a variation from 90 to 360 r.p.m. on the rolls which are 16 in. diameter.

The duty cycle calls for acceleration of the rolls in four seconds to 1380 ft. per minute without load and a stopping with 90 lbs. of plate per roll from 1380 ft. per minute in six seconds, obtained by reducing the frequency from 55 cycles to 12.5 cycles and then plugging to rest. Two such cycles per minute continuously are to be met.

The use of the motor reducer unit in this case has greatly reduced the size of the individual drives and represents the most highly efficient drive that could be obtained for the purpose.

An installation of an electric regenerative drag on a sheet mill has been made at a steel plant in the Detroit district. A 100 hp., 720 r.p.m., synchronous speed, wound rotor induction motor is geared to the driven roll at the end of the roll train so that at all ordinary mill speeds, the motor operates above synchronous speed. Therefore, when its stator is connected to a power supply circuit, it acts as an induction generator. The output is automatically controlled by varying the resistance in the rotor circuit by means of a control system consisting of contactors and notching back relays. The electric drag not only permits of accurately adjusting and maintaining the desired dragging torque, but also brings about a very considerable power saving as the generator output is returned to the power system. With the ordinary friction drag, the dragging power is simply wasted as heat.

During the year there has been developed and built an ingenious feeding device for hot and cold saws. A 72 in. hot saw, for instance, was driven by a 300 hp., a-c., induction motor, 720 r.p.m. Connected to an extension of the armature on this motor by means of a flexible coupling was a 42 kw. generator, which supplied current for a 35 hp. variable speed motor which actuated through a spur gear reduction unit the feeding and return strokes of the saw carriage. For the feeding stroke this motor received its current from the generator mentioned above, and for the return stroke it received its current from a shop circuit. This system of electrical arrangement is known as the indirect type, wherein a variable voltage is used in the feeding and constant voltage is used in the return stroke. This arrangement allows the operator to control the feed so as to obtain maximum production, and permits a rapid return stroke of the saw carriage.

CONTROL EQUIPMENT

For many years photo-electric cells have been serving an extremely useful mission in experimental and industrial work, in controlling the operation of apparatus and equipment, in controlling various processes and in many counting operations. A number of different type cells have been available for experimental, commercial and scientific applications, each type, however, had its own inherent limitations in operating performance or had to be used in conjunction with expensive auxiliary equipment.

Electronic in its character, the new photronic cell employs a highly light sensitive disc which transforms light energy directly into electrical energy without the use of any auxiliary voltage whatsoever. Its response to light variations is instantaneous and sufficient current is developed to directly operate relays without the use of auxiliary apparatus or any battery.

As far as is known, the life of the cell is practically unlimited. Since it does not contain any liquid nor require vacuum or gas, there is nothing to get out of order. It is not subject to physical or chemical change and it has a constant output. It can be exposed to direct sunlight without deterioration, has no dark current, and is non-microphonic.

A photronic illumination control has been developed which consists of two parts: a light collector and relay cabinet. The light collector is a box about six inches in diameter which contains three photronic cells. It has a glass window that exposes the cells to the light and the whole box is weather-proof.

A time delay relay is in the circuit between the sensitive relay and the power relay so that the contacts will not close when clouds pass over the sun and cause the lights to flash on and off. This time delay relay has a time element of about one minute which is ample to take care of the slight fluctuations in light intensity that occur during the hours of daylight.

The lead resistance between the cell box and the relay cabinet does not effect the correct operation of the equipment and permits the cell box to be located on the roof of the building and the relay cabinet to be located inside near the circuits that are to be controlled.

Although the above description applies to the equipment for the control of outdoor lighting the photronic illumination control relay will function also for the control of interior illumination.

The first of the species, the grid glow, could handle directly less than seven watts, the power of a doorbell. A new power grid has been developed which can handle 900,000 watts, 180 amperes at 5000 volts — the power of a locomotive or lake freighter of the smaller sort. It is extremely sensitive to control, extremely efficient.

Industry will find extraordinary uses for such a tool. Already theater lighting is controlled by grid-glow tubes, with very large savings. Their use for electric welding and the control of motors is plainly in sight. In the distance are possibilities of new systems of power transmission.

An automatic screwdown control equipment was developed which permits the roller to set up a schedule of passes on a master control board, then as the operator rolls an ingot or plate, he simply presses a button or advances a master switch one notch after each pass, and the top roll quickly and automatically is moved to a position corresponding to the set-up on the master board for that particular pass.

A new type of speed regulator was developed for direct-current motors in steel mill service which provides great sensitivity and a wide range in adjustment. Essentially, it consists of a balanced-voltage vibrating relay but differs from the conventional type in that it has both movable and stationary coils instead of the usual stationary coil and movable core.

One of the leading control manufacturers has placed on the market a dynamic braking bridge-stop control. This equipment is used in connection with the bridge motion controller of electric cranes and automatically provides a quick stop by dynamic braking upon power failure, due to loss of voltage, tripping of circuit breakers or similar causes. It operates in both directions of travel regardless of the position of the controller handle. It can also be arranged to operate automatically by track type limit switches as the crane enters the danger zone at each end of travel.

The bridge-stop control was developed to provide additional safety for roller bearing cranes where the free running speeds reach values greatly in excess of those previously encountered on cranes not equipped with anti-friction bearings. This control also limits the free running speeds to safe values and the control can also be used for service braking, relieving the foot operated brakes of the work that they are ordinarily called upon to do.

The dynamic braking bridge-stop control is also suitable for trolley motions of ore and coal bridges and various man trolley cranes such as soaking pit cranes, charging machines, also transfer and larry cars, etc.

An automatic discharge magnet control has been perfected during the past year. This is an improved lifting magnet control device with two position master "on" and "off". It is not necessary to return the master from the "off" position to a neutral position when the magnet discharges its load, as the control device automatically connects discharge resistor across the magnet terminals, reverses the current through the magnet, and holds the reverse current on for the correct length of time necessary to release the load. This time will vary with loads of different characteristics.

This control device simplifies the duties of the crane operator and increases the average lifting capacity of the magnet due to eliminating any unnecessary heating that might be caused by leaving the reverse current circuit closed any longer than is necessary to clean the magnet.

CRANES

The steel industry has spent considerable time this year in preparing for the return of good times. Ore will be needed again in great quantities by blast furnaces. Recently in the Pittsburgh district one of the country's most modern ore bridges has been installed ready to handle it.

The rails on either side of the pit are 190 ft. apart. The bucket takes 17 tons (34,000 lbs.) at a bite, enough to load three motor trucks of the biggest size on the highways. Ordinary sizes for such buckets are 6 to 12 tons.

The most advanced feature of this ore bridge is its unusual use of compressed air. Its ten motors, 275 to 7.5 hp., are controlled electro-pneumatically, like a subway train, an arrangement which requires less space and less repairs. Air-released, spring-set brakes for quick, positive stopping are fitted to most of the motors. Because the bridge is so large, there are three sets of compressors and reservoirs, one in the operating cab and one on each leg.

The bridge in normal service can handle 1000 long tons an hour, and feeds four blast furnaces. The loaded bucket rises at 235 ft. per min., the trolley moves along the bridge at 800-900 ft. per min., and the entire bridge can rumble down the track at 175 ft. per min.

There seems to be a belief that welding has so established itself in the steel mill industry that it now needs very little sales or engineering promotion, and that the crane specifications which will be written in future years will be drawn very largely around welding.

To show to what extent the use of welding in crane building has grown, one manufacturer of electric cranes advises that they have built and placed in operation over 600 crane units consisting of one piece trolley frames, one piece end trucks and completely welded girders. They also add that they have not had one failure in this large number of manufactured units.

One of the new developments recently added to the crane equipment is the hydraulic internal expanding type brake for bridge motion, mounted directly on the motor armature. This brake is designed as a self-contained unit, the shoes and all operating parts being housed by the brake wheel and face plate to which they are attached. Owing to the construction of the brake, any wear on the shaft or bearing will not effect the adjustment because of the fact that the position of the brake shoes, relative to the brake wheel, is maintained independently by the ball bearing on the hub of the brake wheel. The master cylinder, also the cylinders which operate the brake shoes, are standard Lockheed parts.

There has also been developed and installed a new bridge drive in which the motor jack shaft is eliminated and both ends of the armature extended through flexible couplings to oil tight gear boxes, one located near either end of the crane. Each of these gear boxes contain double gear reductions and are mounted either directly to the girder or on suitable structural brackets projecting from the girders. Connection from these gear boxes to the driven track wheel axles are made through flexible couplings and a short shaft, this shaft being of the proper length to take care of any angularity or offset misalignment which may result from imperfect runways, etc.

Heretofore where worm gear reduction units were used, especially on cranes, the arrangement was such that the worm could not be removed without disturbing other parts. The solid worm unit housing used heretofore necessitated the removal of the entire reduction unit in case a worm had to be removed for inspection or repairs. There has been developed a worm reduction unit housing wherein the worm enclosure is provided with a cap which can readily be removed, allowing the worm to be lifted out without disturbing any other parts. The worm housing has been greatly improved by an ingenious arrangement of baffles, which distribute ample oil to the worm and to the worm bearings at each end, and with this arrangement the oil level in the worm bearings is maintained at a constant level.

Ingot strippers have been developed this year which will strip three distinct types of ingots: first, the small-end-up or regular type mold; second, big-end-up with hot top; third, big-end-up with sink-head, wherein the top of ingot is flush with top of mold. This latter type of mold requires a special type of stool, which is very simple in design and easy to cast.

MISCELLANEOUS

There appeared this year a new device to defeat lightning's attacks on transmission lines. It is, in effect, merely a length of small-bore fiber tubing with an electrode in each end. The usual strings of insulators, with this tube connected in parallel, become virtually proof against flashover. When, as a result of lightning or other cause, a dangerous surge of over-voltage comes along, it flashes over inside the tube to ground, saving the insulator string. The arc inside the tube is quenched at the first half cycle by the de-ionizing effect of the tube's walls, and the customers whom the transmission line is serving, far away, need not observe even a flicker in their lights.

Flashover protectors, being simple, are inexpensive, and can be placed economically at every insulator on the line. A 50-mile mid-western transmission line is now so equipped.

The thrustor, an electro-hydraulic device for obtaining a straight line motion, is being used in considerable numbers by the steel industry. Among various applications are the operation of shear stops; the opening of furnace doors; operating tilting and lifting tables; operating brakes on skip hoists. Six large thrustors, each capable of developing 4000 lb. thrust through an 18 in. stroke, are just being installed in a Pittsburgh steel plant for operating the doors on billet heating furnaces. Thrustor operated valves are being used for automatically controlling high pressure water on strip and other mills. The thrustor appears to be filling a long felt want for some simple means of obtaining a straight line motion without making use of complicated gearing, limit switches or overload protection.

During special tests this year, a 1500 ampere, 15,000 volt, high-speed de-ion circuit breaker in five hours performed the equivalent of many years' actual service. Seventy-two short-circuits were thrown on it as fast as laboratory connections and oscillograph films could be changed. At 11,000 volts, it interrupted currents from 2720 to 50,500 amperes, (60 cycles) and from 10,000 to 23,700 amperes (25 cycles), all the laboratory could supply at the time. No short-circuit lasted more than 0.04 sec.; some only 0.024 sec. No repairs or inspections were given the breaker, and after the test, which was the most severe ever reported having been given any breaker, it was still operable, and stood a potential test of 50,000 volts.

De-ion Grids, adopted last year to increase the efficacy of high-voltage breakers, including 230 kv. went this year into high current ratings at lower voltages—34.5, 23, and 15 kv. Most interesting are the twenty-eight 15 kv. units for generator and bus tie service at Safe Harbor, which are rated at 2000 to 4000 amperes, and can rupture 60,000 amperes—a million and a half kv-a. The highest previous de-ion current rating was 1200 amperes. The main current is carried by copper brushes especially shaped to avoid sharp turns in the current path. The high magnetic pressures force the contacts together, rather than apart. Compared to ordinary breakers, short circuit currents are ruptured with a reduction of 50% of arc energy dissipated, and a corresponding reduction in pressure, oil deterioration, and maintenance.

The developments in air circuit breakers during the past year have been largely characterized by the increasing trend to steel enclosed air circuit breaker equipment, with the resulting requirements for larger air circuit breakers of the enclosed type. The most noted advance in this line was the development of a truck type air circuit breaker, rated 4000 amperes, 600 volts, 3 phase, 60 cycles at 20 degrees rise above the air outside the cell.

Increasing interrupting capacity requirements have caused developments which minimizes the arcing time, and the form of arcing tip construction which replaces the old familiar phosphor bronze spring type of carbon support with a type of construction in which the contact pressure of the carbon and shunt blocks are maintained by steel coil springs, and the mechanical form of which is such as not to be disrupted by the magnetic forces caused by opening severe short circuits.

Another development by one circuit breaker manufacturer is the silver facing of contact blocks and laminated bridges, giving the advantages of a laminated contact plus the contact maintaining quality of silver.

In the small circuit breaker line one of the most important developments is a "dual overload", which consists of a long time limit normal current armature and an instantaneous armature set at ten times normal current, combined with the same overload magnet. This feature is used in the small circuit breakers for a-c. motor starting duty. The long time limit permits starting heavy loads with low current setting while the instantaneous armature opens the breaker immediately on short circuits.

The advent of the new photronic light sensitive cell has made it possible to offer to illuminating engineers a new, unique, portable illuminometer.

Measurement of light intensities with this new foot-candle meter is as simple and direct a procedure as the measurement of voltage in a circuit with a voltmeter. It is only necessary to turn the searching unit to the light to be measured and then read its value in foot-candles directly on the instrument scale.

There are a great many uses in industry for a means of obtaining an indication or record of values measured at distant points.

A comparatively simple remote metering device has been developed. It consists of three co-operating units: A means of measuring the values to be transmitted, the primary meter. A means of producing a small direct current which is at all times proportional to the quantity measured by the primary meter. A recording instrument which produces a record corresponding to the direct-current supplied by the transmitter.

The primary meter may be a wattmeter, voltmeter, pressure or power factor meter, a water level measuring device or any one of a large number of measuring devices. The only requirement is that it have a member which takes a position corresponding to the value measured.

A transmitter function to produce a direct-current whose value at all time corresponds accurately with the position of this pointer controlled by the primary unit. This requires two things, the production of a direct-current and the control of its volume to correspond to the indication to be transmitted.

Both of these functions, the production and control of the transmission current, are accomplished in a simple manner by the use of ordinary three-element vacuum tubes.

During the year, testing machines were in substantially continuous operation, determining the strength of steel at elevated temperatures.

Under continued stress at elevated temperatures, steel yields continuously. There is no dividing line of temperature above which all steels "flow" or "creep" and below which they do not, but above 400 deg. C. the "creep strength" becomes a vital factor in design. The rate of flow depends on the composition and treatment of the steel, upon the temperature and upon the magnitude of the stress.

In the test, specimens were held continuously and accurately at certain temperatures (400, 450, 500, and 550 deg. C.) for periods of two months or more under known tensile stresses of several thousand pounds per square inch, and the elongation of the specimens was measured from day to day and plotted. The accuracy of these measurements is such that elongation at the rate of one ten-millionth of an inch per inch of length per hour could be observed with reasonable certainty.

The object of these flow tests was to render possible the prediction of the relative ability of materials to maintain dimensions under stress at elevated temperatures over long periods of time, say 10 or 20 years.

Specimens of test pieces are then plotted in stress for flow rate of 1% in 100,000 hours at various temperatures.

There apparently is no voltage too high for use in x-ray or cathode-ray work. By cascading, or applying the total voltage in a series of steps, any potential that can be generated may be used. There was installed a two section x-ray tube operating at 900,000 volts and five milliamperes of current; and a three section cathode-ray tube operating at the same voltage was also constructed for laboratory use. Both of these tubes operate continuously at a voltage far higher than any attained in a single-section tube.

In the industrial field, high-voltage x-ray equipment was used for radiographic examination of thick sections of metal.

Hidden defects in castings, welds, and electric machinery, hitherto difficult or impossible to detect visually or by ordinary methods, are now revealed by the spotlight of the x-ray in the largest laboratory of its kind in industry. This laboratory is completely equipped with a sub-station, rooms for taking x-ray pictures, photographic darkroom, and air conditioning plant.

A 200,000 volt x-ray tube is used, the highest voltage rating at present commercially available, the equipment being of the standard hospital deeptherapy type. Products varying in size from very small devices to castings weighing many tons can be photographed.

In the past, common practice on disconnecting switches has been to provide a large contact area between the switch blade and the switch jaw when the switch was to carry heavy currents. These contact areas have been under a unit pressure of only a few pounds per square inch and were ordinarily figured to carry from 50 to 80 amperes per square inch.

A new switch has been developed which makes contact on a very limited area and with very high pressure per square inch. The pressure is obtained by backing the contacts with heavy phosphor bronze springs and the contact is made on spots raised on the surface of the blade.

With these high pressures it is naturally assumed that the switch would be hard to operate. This is not the case as a simple cam action releases the pressure before the blades are withdrawn from the jaws. With this action a 3000 ampere switch is easily operated with one hand regardless of the condition under which it may be installed.

Perhaps the outstanding development in the past few years in the field of asbestos wires and cables has been the development of the asbestos varnish cambric construction. The insulation here consists of some layers of varnished cambric sandwiched between walls of compactly felted asbestos with a heavy outer asbestos braid. The entire insulation is impregnated so as to withstand normal conduit service conditions. This cable was originally developed for 600 volt service; but the successful results obtained in service have justified increasing the voltage range of this cable so that it can now be supplied up to and including 7500 volts. Another development is a new type of black finish which is particularly applicable to all types of cable having an asbestos braid. The new finish consists of a special insulating compound which, when used for impregnating the outer braid and then polished by a special machine, gives the cable a high jet black lustre, filling all the interstices in the braid. This new finish not only adds to the cables appearance but makes for ease in pulling through conduits; gives added moisture resisting qualities, and generally improves the insulation efficiency.

The development of new types of insulating machines, makes it possible to supply Rockbestos round magnet wire in sizes from 15 to 28 inclusive, with no greater overall diameter than double cotton covered wire. This means that the superior insulating qualities of asbestos need not be sacrificed for want of proper space, or the excessive expense of redesigning. For with this new type of small diameter magnet wire, the correct number of turns may be placed in an allotted space without difficulty. In general, asbestos magnet wire can now be supplied in practically all sizes of the round, square, and rectangular, to the same dimensions as double cotton covered wire.

CONCLUSION

Each year the task of preparing a report on Electrical Developments becomes a greater one. No longer can one engineer keep in touch with the details of the development in all lines of the Electrical Industry. We are living in the age of specialists. For this reason, there are some 65 members on this Electrical Developments Committee.

Many of the members of this committee have served in this capacity for years and have contributed largely to the success of these reports. And now each year I look forward with pleasure to the time when letters containing these reports from the members of this committee will be received.

It is not possible to publish all the data submitted nor is it possible for the chairman to know in every case what is of most interest to the members of the A. I. & S. E. E.

The spirit of co-operation, as exemplified by the members of this committee, has been an inspiration to me as Chairman of the Electrical Developments Committee.

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The Behavior of Steel at Forging Temperatures With Respect to Scaling Losses

▲

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To be presented before A. I. & S. E. E., Combustion Engineering Division, at the Annual Convention in Pittsburgh, June 20-23, 1932.

The scaling of steel, that is, its oxidation at elevated temperatures, is a more serious problem than is often appreciated. In the past it has been regarded as a necessary evil and so dismissed with little further thought.

The economic loss actually entailed as a consequence of scaling may be realized by reference to Table I. This table is based on the results of the research reported in this paper applied to steel produced in the mill and to the steel used in forging for the years 1929 and 1931.

It will be seen that for the peak year, 1929, the losses in the steel mill occurring during the soaking heat prior to rolling might easily amount to 344,000,000 to 545,000,000 pounds representing values from $830,000 to $1,310,000. Even during 1929, these losses quite probably amounted to 157,000,000 to 250,000,000 pounds of metal having an intrinsic value of $326,000 to $520,000. In the forging industry alone the losses ranged from 9,150,000 to 14,000,000 pounds for 1929, an economic loss of $220,000 to $352,000, while for the year 1931, the loss of metal quite probably ranged from 3,380,000 to 5,390,000 pounds having a value of $70,000 to $112,000. These figures will serve to indicate the magnitude and value of the amount of metal lost.

The intrinsic value of the metal lost, is however, only one of the detriments of excessive scaling. Scale reacts with most furnace refractories to form a relatively low melting compound which may greatly shorten the life of the furnace. Scale also occupies space in forging dies and may easily result in wrongly dimensioned parts. Should it be pounded or rolled into the metal surface, imperfections result which may lead to failure of the forging or rejection of the steel. Finally, the removal of scale in the finishing operation naturally involves additional labor and equipment costs which increase with the depth and adherence of the scale. It seems evident that this problem of oxidation of steel entails a total cost to the producer considerably in excess of the intrinsic value of the metal lost, a cost which is influenced by factors whose magnitude it is exceedingly difficult to estimate.

From the table it is also evident that both atmosphere and temperature have an important effect on scaling losses. The composition of the atmosphere in the furnace must, therefore, be subject to accurate control. In this respect gas is especially satisfactory. Since it is the most highly refined fuel, the control of the composition of atmospheres formed by burning it with air is more easily effected than with other fuels. Gas heat has, to be sure, other inherent advantages from the point of view of speed and economy. In many applications the high rates of heating capable of attainment in the gas-fired furnace are of considerable advantage in shortening heating times and thus increasing furnace capacity. Gas is also a very economical fuel when used intelligently in furnaces of proper design, but

TABLE I

Partial Estimates of Scaling Losses in Industry

		Losses based on 1929 production		Losses based on 1931 production	
	Condition of atmosphere	Lbs. metal	Value in dollars	Lbs. metal	Value in dollars
Losses in steel mill one heating at 2300°F.	4%O_2	419,000,000	1,010,000	192,000,000	398,000
	1%O_2	388,000,000	935,000	178,000,000	370,000
	2%CO	344,000,000	830,000	157,000,000	326,000
Losses in steel mill one heating at 2400°F.	4%O_2	545,000,000	1,310,000	250,000,000	520,000
	2%CO	449,000,000	1,200,000	205,000,000	425,000
Losses in steel forged one heating at 2300°F.	4%O_2	11,100,000	268,000	4,100,000	85,000
	1%O_2	10,300,000	248,000	3,820,000	79,200
	2%CO	9,150,000	220,000	3,380,000	70,000
Losses in steel forged one heating at 2400°F.	4%O_2	14,600,000	352,000	5,390,000	112,000
	2%CO	12,000,000	289,000	4,400,000	91,100

its most important claim for consideration in heating furnaces lies in the fact that with gas heat, it is relatively easy to control the atmosphere and temperature.

The importance of this problem of scaling and the fact that information regarding it was so limited prompted the Committee on Industrial Gas Research of the American Gas Association, about three years ago, to institute and carry on a program of scaling research at the University of Michigan. It was realized that an accurate knowledge of how metals will behave under any given conditions of operation is essential to the proper design and operation of a furnace for metallurgical application. This paper presents the results of that research.

The method of investigation used consisted essentially in exposing small steel samples to the action of the chosen atmosphere under the desired conditions of velocity and temperature for given periods of time, quenching, stripping, and determining the loss in weight. The samples were heated in the small gas-fired furnace shown in Figure 1, which had an outer carborundum muffle and a porcelain tube as an inner muffle. The porcelain tube was used in order that the furnace atmosphere might not interfere with the study of the isolated factors. The composition of the various steels studied is given in Table II.

TABLE II

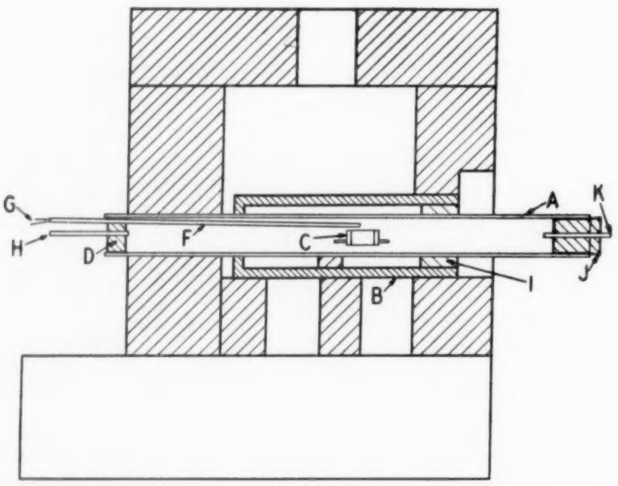
Chemical Composition of the Steels

Steel	%C	%Mn	%Si	%S	%P	%Ni	%Cr	%Va	%Mo	%W
Electric Iron	.012	.008	.001	.007	.002	......	......	......	......	......
1015	.19	.53	.18	.020	.016	......	......	......	......	......
1030	.28	.653	.01	.032	.013	......	......	......	......	......
1050	.52	.64	......	.034	.025	......	......	......	......	......
1090	.88	.34	......	.018	.029	......	......	......	......	......
High C. Tool	1.12	.33	.21	.026	.011	......	......	......	......	......
2320	.17	.59	.24	.020	.018	3.40	......	......	......	......
3130	.35	.56	......	.025	.016	1.11	.56	......	......	......
6145	.47	.58	.16	.015	.015	......	.98	.24	......	......
4140	.39	.67	......	.024	.020	......	.93	......	.32	......
4615	.21	.54	.29	.012	.017	1.81	......	......	.25	......
High Speed	.73	.32	.08	.018	.025	......	4.27	1.06	......	18.62
Krupp	.12	.45	.25	.03 max.	.03 max.	4.00	1.50	......	......	......
KA2	.08	......	......	......	......	9.00	17.50	......	......	......
Nitralloy	.35	.45	.30 max.	.025 max.	.025 max.	......	1.25	......	.20	1.00 Al

TABLE III

Effect of Variation of the Size of Sample on Scaling of S.A.E. 1015 Steel in Dry Air at 2300° F.

Sample Number	Total Time in Furnace (Minutes)	Rate of Flow Ft. per Minute	Diameter of Sample (Inches)	% Loss in Weight	Loss in Lbs. per 100 sq. in.	Wt. of Scale Lbs. per 100 sq. in.	Av. Loss in Wt. Av. Wt. of Scale
AR7	40	60	.500	7.96	.123	.171	
AR8	40	60	.500	7.58	.118	.166	
Average				7.77	.121	.169	.715
AR9	40	60	.375	9.65	.115	.169	
AR10	40	60	.375	9.99	.123	....	
Average				9.82	.119	.169	.709
AR11	40	60	.6197	6.86	.129	.180	
AR12	40	60	.6230	6.66	.126	.174	
Average			.6214	6.76	.127	.177	.719
630	40	60	.8138	4.88	.118	.163	
646	40	60	.8135	5.18	.122	.169	
Average			.8136	5.03	.120	.166	.720



A - Porcelain tube
B - Carborundum muffle
C - Sample
D - Fire brick plug
F - Thermocouple protection tube thermocouple inside
G - Thermocouple leads
H - Air inlet from flow meter
I - Front muffle brick
J - Brick to close tube
K - Gas sampling tube to siphon sampler

FIG. 1 GAS FURNACE USED IN SCALING TESTS

THE SCALING OF STEEL IN SIMPLE GASES

The object of studying the scaling of steel in simple gases was to evaluate the fundamental laws governing the process. The oxidation of steel is essentially a surface reaction; it would therefore be expected that scaling losses for any given steel would be a function of the area of metal exposed. The truth of this assumption is borne out by the figures of Table III which summarize the results of tests on various sizes of S. A. E. 1015 steel at 2300°F. in dry air. The percentage loss in weight increases with decreasing size of sample, but when the loss is expressed in terms of weight per unit area, these differences between samples of varying size disappear. All results are, for this reason, expressed in terms of weight per unit area, the unit of area being chosen as 100 square inches.

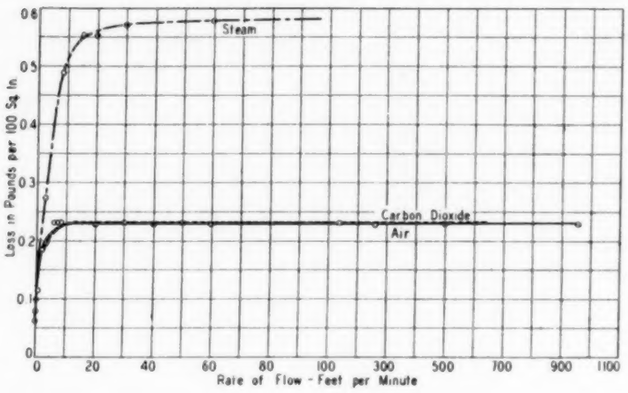


FIG. 2 EFFECT OF VARIATION OF RATE OF FLOW ON THE SCALING OF S.A.E. 1015 STEEL AT 2300°F.

The effect of varying the velocity of the gas stream on the scaling of S. A. E. 1015 steel at 2300°F. is shown in Figure 2. In each case the data indicate that, beginning with rates of flow in the neighborhood of zero feet per minute, there is a rapid increase in scaling losses with increasing velocity. There seems to be however, a critical rate of supply of the oxidizing gas beyond which an increase in the velocity of the gas has no effect, as is shown in the flat portion of the curves. In the case of water vapor the effect of increasing the

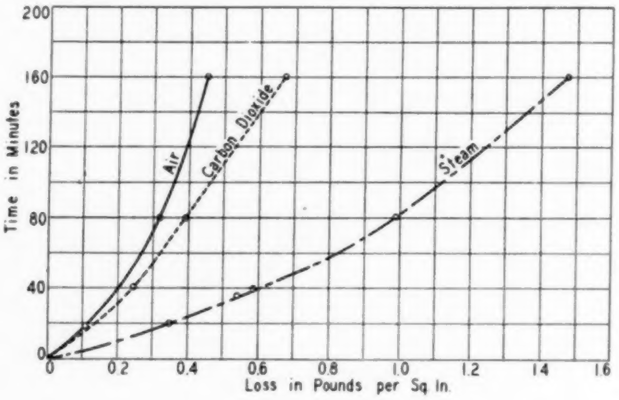


FIG. 3 EFFECT OF VARIATION OF TIME ON THE SCALING OF S.A.E. STEEL AT 2300° F.

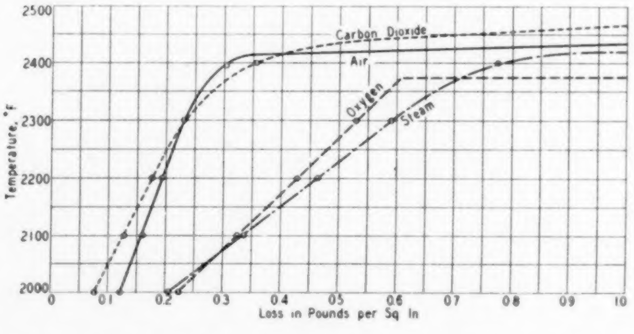


FIG. 4 EFFECT OF VARIATION OF TEMPERATURE ON THE SCALING OF S.A.E. 1015 STEEL

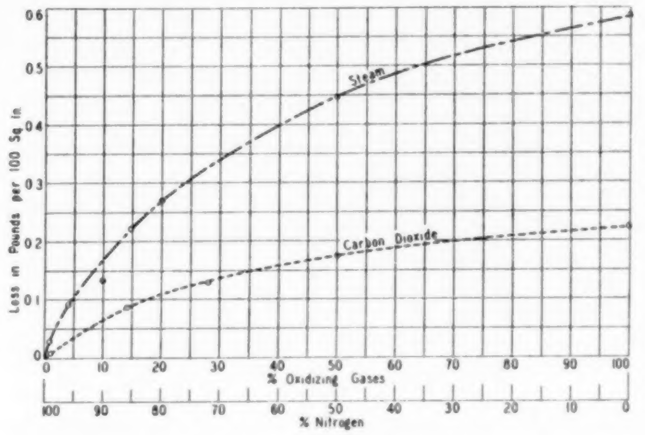


FIG 5 EFFECT OF DILUTION OF OXIDIZING GAS WITH NITROGEN ON THE SCALING OF S.A.E. STEEL AT 2300° F

velocity is more gradual than in either of the other two cases.

Figure 3 shows the effect of varying the time of exposure of S. A. E. 1015 steel at 2300°F. The scaling losses follow a smooth curve, increasing with increasing time. The form of these curves indicates, however, that the rate of scaling diminishes with increasing time, because of the increasing thickness of scale. The increasing thickness of scale operates to reduce continually the amount of oxidizing agent diffusing inward toward the metal. Consequently, since the supply diminishes steadily, the rate of scaling or formation of new oxides must also diminish.

When the temperature to which the steel is exposed is increased, the scaling losses mount rapidly as is shown in Figure 4. Furthermore, it is strikingly evident that the curves, with the exception of that for oxygen, change direction very rapidly between 2400°F. and 2500°F. This very marked increase in scaling losses is accounted for by the fact that in this temperature range the scale becomes liquid and consequently fails in its protective action. The case of oxygen is somewhat different since here the marked increase in scaling occurs at 2375°F. At this temperature the heat developed by the oxidizing reaction raises the actual temperature of the sample so rapidly that the reaction becomes increasingly vigorous, and culminates in the complete burning of the steel to form oxide.

If the partial pressure of the oxidizing gas in mixtures with nitrogen is increased, the scaling losses increase in regular fashion as shown in Figure 5. This result is to be expected, since the greater the partial pressure of the gas the greater the supply of oxidizing agent. This enables more metal to enter the reaction in much the same fashion as was found when the velocity was increased.

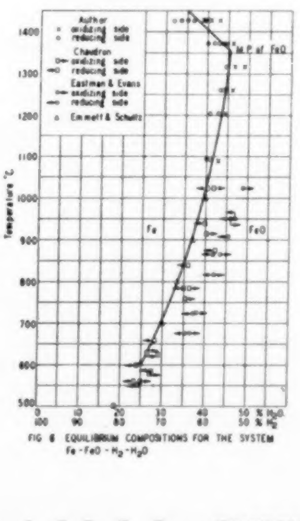


FIG 6 EQUILIBRIUM COMPOSITIONS FOR THE SYSTEM Fe-FeO-H_2-H_2O

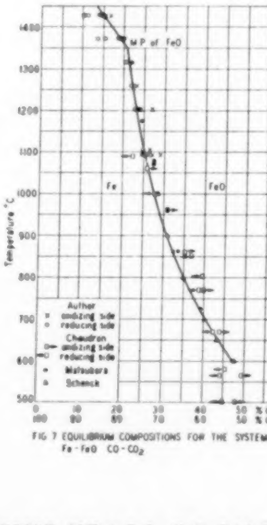


FIG 7 EQUILIBRIUM COMPOSITIONS FOR THE SYSTEM Fe-FeO CO-CO_2

In the forging range of temperatures, 2000°F. to 2500°F., this investigation has shown that steam is the most active scaling agent, while carbon dioxide and air are approximately the same in scaling power. Oxygen does not exceed pure steam in effect until 2375°F. is reached, at which point the oxygen reaction is markedly accelerated.

The fundamental laws resulting from this study might be briefly summarized as follows. First, scaling losses are, at least in most cases, strictly a function of area exposed. Second, scaling losses are increased by increasing the velocity of gas, but attain a constant value for velocities of 10 or 20 feet per minute and higher. Third, scaling increases with increasing time, but does so at a diminishing rate. Fourth, oxidation increases with increasing temperature, and becomes very great when the melting point of the oxides, about 2480°F. is exceeded. Fifth, scaling losses increase with increasing partial pressure of the oxidizing gas.

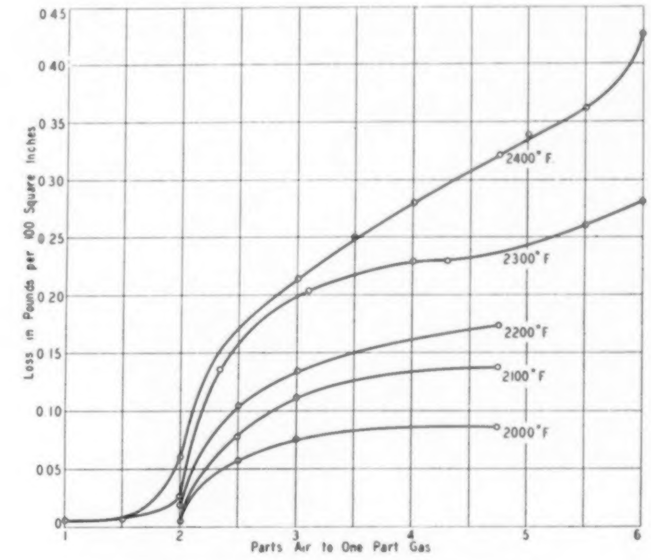


FIG 8 SCALING OF S.A.E. 1015 STEEL IN 40 MINUTES TIME IN ATMOSPHERES FORMED BY BURNING GAS AND AIR IN VARIOUS PROPORTIONS

OXIDATION EQUILIBRIA

The oxidation equilibria chiefly involved in the scaling of steel in furnace atmospheres are

$$FeO + H_2 = Fe + H_2O \text{ and}$$
$$FeO + CO = Fe + CO_2.$$

In this investigation of these equilibria above 2000°F., the amount of scaling produced by any given mixture of hydrogen and water vapor, or of carbon monoxide and dioxide was of no import, since the only knowledge required of any particular mixture was whether or not it produced scaling at the temperature in question. Hence a series of tests were made at various temperatures at 2300°F. for example with mixtures of hydrogen and water vapor containing various amounts of water vapor. These mixtures were designated as scaling or non-scaling, depending upon their reaction upon iron. The limits between the lowest scaling and highest non-scaling mixtures were from 0.4 per cent to 2.0 per cent of water vapor. This procedure was repeated at several temperatures between 2000° F. and 2600°F. The results are plotted in Figure 6, in which the upper portion of the curve is so drawn that it lies at each of the chosen temperatures between the closest scaling and non-scaling compositions. The lower portion of the curve is due to Emmett and Schultz.[1].

Much the same procedure was followed where iron was exposed to mixtures of carbon dioxide and carbon monoxide. The results of this study are shown in Figure 7, where gas composition is plotted against temperature. As in the former case this curve is so drawn as to lie between the closest scaling and non-scaling mixtures at each temperature. The lower portion of this curve is taken from Schenck[2]. It will be noted that in both cases there occurs a sharp break in the curves at 1360°C. (2480°F.) This break occurs because of the melting of ferrous oxide. Consequently if the melting point of ferrous oxide, the chief component of scale, is exceeded, more hydrogen or more carbon monoxide will be required to prevent scaling. Finally, the curves presented in these figures divide all possible gas compositions into two sorts; either scaling, to the right of the curves, or non-scaling, to the left.

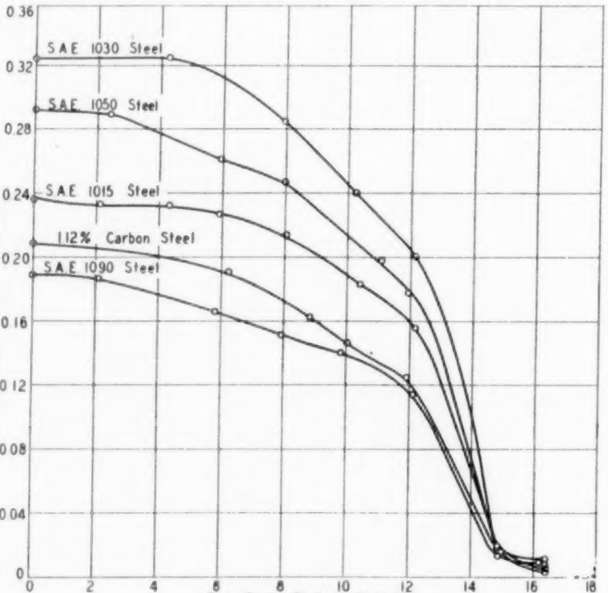


FIG. 9 SCALING OF PLAIN CARBON STEELS AT 2300°F IN 40 MINUTES TIME IN VARIOUS FURNACE ATMOSPHERES AS SHOWN BY THE LOSS IN WEIGHT OF THE STEEL

THE SCALING OF STEEL IN GAS-AIR COMBUSTION ATMOSPHERES

In Figure 8 are shown the results of exposing S. A. E. 1015 steel samples to various combustion atmospheres ranging in ratio from 2-air-to-1-gas to 6-air-to-1-gas at temperatures from 2000°F. to 2400°F. The curves, with the possible exception of that for 2400°F., are strikingly similar. Complete combustion for the particular gas being used was secured with a ratio of 4.75 parts air to 1 part gas. With ratios less than this, that is when excess gas was used, the general tendency is to reduce scaling losses, though this is not marked until the air-gas ratio has been decreased below 3 to 1. Non-scaling conditions are attained with ratios of 2 air to 1 gas and less.

At 2300°F. and 2400°F. the range of ratios was extended to 6 air to 1 gas. In these cases excess air was present and consequently free oxygen was always found in amounts ranging from about 0.5 to 4.0 per cent. It will be observed that the presence of free oxygen in combustion atmospheres leads to markedly increased scaling losses, especially at 2400°F. Such atmospheres as these must of course be avoided in operating furnaces.

The curves of Figure 9 were obtained when various plain carbon steels were subjected to atmospheres containing increasing percentages of carbon monoxide at 2300°F. The curves are very similar, as was to be expected, and show much the same characteristics as those of Figure 8. The reduction of scaling losses is not decidedly marked until considerable excess gas is used, leading of course to increasing percentages of carbon monoxide. The order of scaling, in decreasing magnitude, appears to be as follows: S. A. E., 1030; S. A. E., 1050; S. A. E., 1015; 1.12-per-cent carbon steel; and S. A. E., 1090. Apparently carbon in the steel has but little influence upon the scaling losses. This work was next extended to include several of the more common alloy steels as shown in Figure 10. Here again, with the exception of high speed steel of the high tungsten type, the scaling losses are reduced somewhat gradually as the carbon monoxide is increased from 0 to 6 per cent, then more rapidly with further increases until non-scaling conditions are attained with atmospheres containing 15 to 17 per cent carbon monoxide.

In predicting the effect of any combustion atmosphere upon steel the equilibria discussed under Section II are of inestimable value. When city gas and air are burned, gas being in excess, carbon dioxide, carbon monoxide, water vapor and hydrogen are formed. These then adjust themselves at

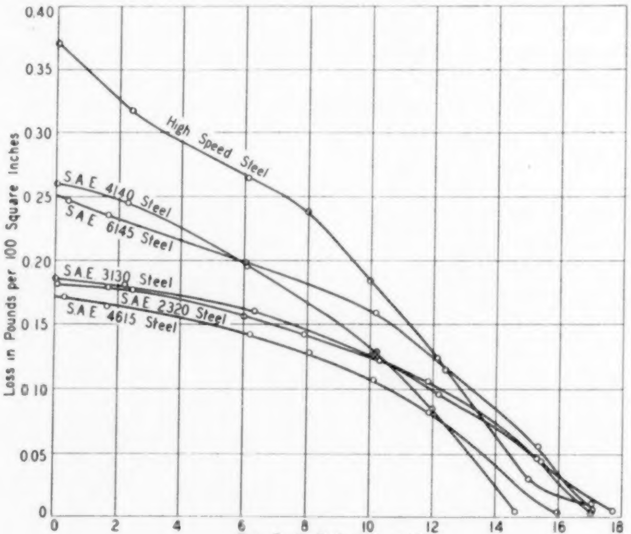


FIG. 10 SCALING OF ALLOY STEELS AT 2300° F IN 40 MINUTES TIME IN VARIOUS FURNACE ATMOSPHERES

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1. J. Am. Chem. Soc. Vol. 52 p. 4268—1930.
2. Z. anorg. allgem. chem. **166**, p. 113, **167**, p. 254, **167**, p. 315.

equilibrium in amount to conform to the water-gas constant for the particular temperature. The water-gas reaction $CO_2 + H_2 = CO + H_2O$ has as its equilibrium constant $K = \frac{[CO]\ [H_2O]}{[CO_2]\ [H_2]}$. This constant may likewise be derived from the reactions with iron;

$$FeO + CO \leftrightarrow Fe + CO_2 \quad K_1 = \frac{\%CO_2}{\%CO}$$

and

$$FeO + H_2 \leftrightarrow Fe + H_2O \quad K_2 = \frac{\%H_2O}{\%H_2}$$

by a simple division of K_2 by K_1. Hence if any atmosphere is to be non-scaling to steel it must have certain maximum ratios of $\frac{CO_2}{CO}$ and $\frac{H_2O}{H_2}$, since the non-scaling compositions must lie to the left of the curves of Figures 6 and 7.

TABLE IV

Comparison of CO_2 — CO and H_2O — H_2 Ratios for Scaling and Non-Scaling Tests on Various Steels at 2300° F.

Run No.	Steel	$\frac{\%CO_2}{\%CO}$ Ratio	$\frac{\%H_2O}{\%H_2}$ Ratio	Condition After Testing
A1-19	1030	.346	.879	Scaled
A1-20	1030	.199	.479	Not scaled
A2-15	1050	.346	.880	Scaled
A2-17	1050	.199	.479	Not scaled
A3-10	1090	.675	1.85	Scaled
A3-16	1090	.199	.479	Not scaled
E-15	1.12%C	.333	.875	Scaled
E-17	1.12%C	.337	.310	Not scaled
FF-14	2320	.450	1.07	Scaled
FF-15	2320	.238	.639	Not scaled
G-14	3130	.431	1.07	Scaled
G-16	3130	.219	.495	Not scaled
Q-21	6145	.420	1.05	Scaled
Q-15	6145	.238	.639	Not scaled
L-11	4615	.610	1.36	Scaled
L-14	4615	.304	.670	Not scaled
KK-14	4140	.530	1.47	Scaled
KK-3	4140	.300	.520	Not scaled
YY-13	High speed	.354	.803	Scaled
YY-15	High speed	.238	.639	Not scaled

Value $\%CO_2/\%CO$ ratio from Figure 5 .288 at 2300°F.

Value $\%H_2O/\%H_2$ ratio from Figure 6 .792 at 2300°F.

For example, the maximum non-scaling ratios of $\frac{\%H_2O}{\%H_2}$ and $\frac{\%CO_2}{\%CO}$ at 2300°F. as determined from Figures 6 and 7 are 0.792 and 0.288 respectively. In Table IV are summarized a few of the data obtained during the investigation in which the atmosphere analyses are expressed as the ratios $\frac{\%CO_2}{\%CO}$ and $\frac{\%H_2O}{\%H_2}$. On the basis of the comparisons made in this table atmospheres whose ratios are in excess of the values 0.288 and 0.792 produce scaling at 2300°F., while those atmospheres whose $\frac{CO_2}{CO}$ and $\frac{H_2O}{H_2}$ ratios are below these values produce no scaling. In general it may be remarked, that at equilibrium, when one of these ratios is below the maximum, the other will be below also, and likewise that, when one is greater in value, the other will be greater. It would be much easier in plant practice, therefore, to use the more easily determined $\frac{CO_2}{CO}$ ratio as the criterion. It must be emphasized, though, that the success of this prediction is necessarily based upon the condition that equilibrium in the furnace atmosphere is attained before the steel surface is reached. Without such a condition any prediction would be unsafe.

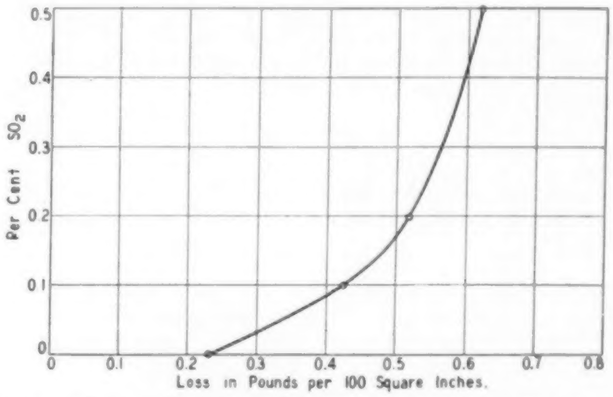

FIG. 11. EFFECT OF VARIATION OF PER CENT SO_2 ON SCALING LOSSES OF S.A.E. 1015 STEEL. ATMOSPHERE - COMPLETE COMBUSTION OF CITY GAS AND AIR.

THE EFFECT OF SMALL AMOUNTS OF SULFUR DIOXIDE in THE PRODUCTS OF COMBUSTION

The presence of 1 to 2 per cent of sulfur in some fuels could lead to sulfur gases in the furnace atmosphere in amounts up to 0.2 per cent. For most operating conditions this sulfur would be present in combustion gases as sulfur dioxide since it is only in the presence of considerable hydrogen and carbon monoxide that less oxidized forms are stable. Small amounts of sulfur dioxide are shown to be extremely powerful in their effect on scaling losses. This is illustrated in Figure 11 where scaling losses of S. A. E. 1015 steel at 2300° F. are plotted against per cent sulfur dioxide contained in atmospheres resulting from complete combustion of gas and air. An amount as small as 0.1 per cent sulfur dioxide nearly doubles the loss, increasing it from about 0.23 pounds per 100 square inches with no sulfur dioxide to 0.43 pounds per

100 square inches with 0.1 per cent. Further increases of sulfur dioxide, corresponding to the use of fuels containing upwards of 1.0 per cent sulfur, cause further increases in scaling losses.

The influence of sulfur dioxide on scaling losses increases rapidly with increasing temperature as shown in the curves of Figure 12, where losses are plotted against temperature for atmospheres containing no sulfur dioxide and for those containing 0.2 per cent. The curves for non-sulfur containing atmospheres and sulfur containing atmospheres become increasingly divergent at higher temperatures. The drastic effect of sulfur bearing atmospheres is likewise clearly evident in the curves of Figure 13 where scaling losses are plotted against time of exposure at 2000°F. and 2300°F. Comparison of the form of these curves with the form of those in Figure 3, in which are given the results with the normal oxidizing components of combustion atmospheres, carbon dioxide and steam, shows that scaling in sulfur bearing atmospheres proceeds at an approximately constant rate, the scale having lost its protective qualities which caused the decreasing rates shown in Figure 3. Some of the alloy steels, particularly those containing nickel, are even less resistant to sulfur bearing atmospheres than the S. A. E. 1015 steel for which results have been shown.

SUMMARY

In conclusion the points which have been discussed in this paper may be summarized in the following:

1. Scaling losses are effected by velocity of the gas stream only in the range of very low velocities, 0 to 10 feet per minute.

2. Scaling losses increase with increasing time, but the rate of scaling in sulfur free atmospheres decreases with increasing time.

3. Scaling losses increase decidedly with increasing temperature, becoming very great when the melting points of the scale, about 2480°F. is exceeded.

4. The effect of any combustion atmosphere upon steel with respect to scaling may be predicted from the equilibria data of the reactions $FeO + CO \leftrightarrow Fe + CO_2$ and $FeO + H_2 \leftrightarrow Fe + H_2O$.

5. Scaling losses in combustion atmospheres are increased by the presence of free oxygen and decreased by the presence of free carbon monoxide, though, to be sure, this decrease is not marked up to 6 or 8 per cent.

6. The presence of as little as 0.1 per cent sulfur dioxide in the products of combustion, seriously increases scaling losses at all temperatures in the forging range and larger amounts increase these losses still more. The results of this investigation have shown unquestionably that control of atmosphere and temperature are of much importance in controlling scaling in the forging range. Gas fired furnaces are the most susceptible to the accurate control of temperature and atmosphere necessary to reduce scaling. On the basis of the results of this research certain recommendations may be made regarding the operation of furnaces designed for use in hot working steel.

1. Free oxygen should be avoided in furnace atmospheres at all times.

2. Operation of most of the existing furnaces with amounts of excess gas sufficiently large to greatly reduce scaling would probably prove uneconomical for most work. It is advisable, however, to operate with a small percentage of carbon monoxide in the furnace atmosphere in order to insure the absence of oxygen.

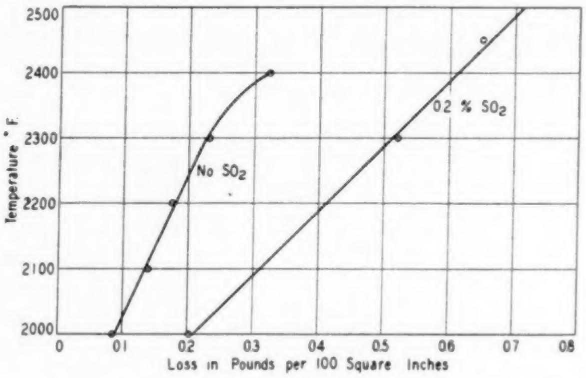

FIG 12 EFFECT OF VARIATION OF TEMPERATURE ON SCALING OF S.A.E. 1015 STEEL IN SULFUR CONTAINING ATMOSPHERES COMPARED WITH NON SULFUR BEARING ATMOSPHERES. ATMOSPHERE COMPLETE COMBUSTION CITY GAS & AIR.

3. The time of exposure of steel to furnace atmospheres should be kept as low as possible.

4. The temperature of exposure should in most cases, be kept well below the melting point of the scale, 2480°F., to avoid undue scaling.

5. Fuels high in sulfur are likely to cause extremely high scaling losses and their use can not be recommended.

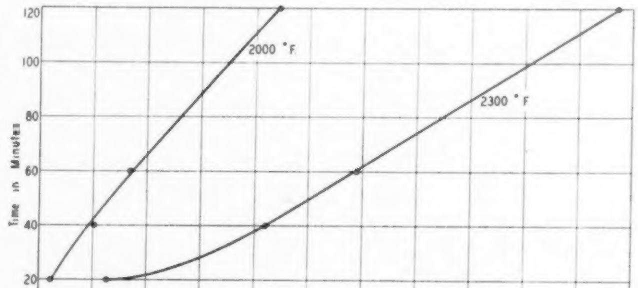

FIG 13 EFFECT OF VARIATION OF TIME OF EXPOSURE TO SULFUR BEARING ATMOSPHERES ON SCALING OF S A E 1015 STEEL ATMOSPHERE COMPLETE COMBUSTION CITY GAS AND AIR

ACKNOWLEDGMENT

The author desires to express his appreciation to the Committee on Industrial Gas Research of the American Gas Association, whose financial support enabled this study to be made.

"Some Operating Experiences on Continuous Lubricating Systems for Main Roll Drives"

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By L. F. COFFIN
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Some interesting papers have been published recently* dealing with both equipment and lubricants for continuous oil circulating systems as applied to heavy duty rolling mill drives. In view of the increasing number of such installations, a few operating experiences from some of these may be worth recording.

In a 27", 24" and 21" Continuous sheet bar and skelp mill rolling skelp up to 22¼" wide, and placed in service in June, 1927, the horizontal stands are driven by 3 main motors of 4000, 6500 and 6700 H.P., each motor driving 4, 4, and 3 stands respectively through herringbone reduction gears entirely encased and oil lubricated. All three sets of gears, mill pinions and gear and pinion bearings and 3 edging mills were originally lubricated with the same oil of the following characteristics:

Gravity	26.5
Cold Test	35° F.
Flash	500° F.
Fire	575° F.
Viscosity at 100° F.	1700 secs.
Viscosity at 210° F.	119 secs.
Demulsibility	1620 secs.

The capacity of each of the three systems is 6000 gallons of oil. The system for the "A" or slow speed drive has 12 gear and pinion sprays and 28 bearing feeds, circulates 35 to 40 gallons per minute and has an operating temperature of 105° F. in summer and of 80° F. in winter. The system for the "B" or medium speed drive has the same number of feeds, circulates from 40 to 45 gallons per minute and operates at about 110° F. in the summer and 90° F. in the winter. The system for "C" or high speed drive has 6 gear sprays and 22 bearing feeds and circulates 50 to 55 gallons per minute and operates at 120° F. in summer and 100° F. in winter.

The three vertical edging mills originally were fed from these systems and returned their oil to them, but difficulty was experienced from the start in keeping water out of the edging mill gear cases until finally the oil in the entire "B" drive became so badly emulsified in spite of continual centrifuging that it was necessary to shut down and replace all the oil in this drive. At the same time, a separate circulating system for the 3 edging mills was installed and a lighter oil used in these edgers of 210 sec. viscosity. This oil is more easily separated from water. Another source of water and dirt contamination was in the drains from the mill pinion bearings next to the mill due to water from coupling boxes on spindles and from hydraulic sprays. This oil is now returned to a separate tank, heated, settled and then pumped into the main drain line. This experience and others led to the practice of using as light an oil as possible on each drive in order to get the maximum separation from water and dirt. Oils of 500 viscosity and of 1700 viscosity may both have the same demulsibility factor of 1620 but actually the former oil is far safer and easier to control where there is any dirt or water contamination. The oil in the "B" drive has been reduced by adding light turbine oil of 150 viscosity so that the resultant viscosity in the whole system is now 1000 seconds. In the "C" drive, the viscosity has also been reduced to 1000 seconds. Gear and pinion wear since reducing the viscosity has been no more noticeable than when all the systems operated at 1700 seconds. The following tables of thickness of gear and pinion teeth showing wear are given on page 268.

Total tonnage rolled 9-2-1928 to 1-31-1932 was 1,110,580 tons. A Brown & Sharpe tooth gauge was used for measuring tooth thickness, but too much reliance cannot be placed on the actual wear in any one case as the same tooth could not always be measured in each case. The average wear, however, as indicated can probably be relied on as being approximately correct and shows .007" for the main reduction gears, .024" for the main motor pinions and .018" for the mill pinions for the period given. Oil samples are taken at regular intervals and tested for water, dirt and viscosity and tested every six months for saponifiable matter. Oil is centrifuged in a vertical bowl three wing type 350 gallon per hour portable centrifuge when water shows above .50% or dirt above .10%. On a 6000 gallon system of 1700 visc. oil a complete cleaning requires 400 to 500 hrs. Experiments on 55 gals. of emulsified 1700 visc. oil show that running oil through centrifuge at rate of 300 gallons per hour for 6 or 7 hours, will not clarify it, but when run 3 hours and then the bowl on centrifuge is cleaned, the oil will clear up in 30 minutes. A temperature of 150° F. when starting to centrifuge and then, after danger of foaming is passed, a temperature of 180° F. seems to give good results on this oil.

All three of these reduction gear units are equipped with overflow seals so that gears may be run in bath as well as with sprays, but experience has shown that some of these seals, especially on the higher peripheral speed gears must be by-passed at all times and that in starting in cold weather, all must be by-passed until cases warm up to about

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* "Modern Oil-Lubrication Methods as Applied to Rolling Mill Equipment," by Joseph A. Merrill and Edwin M. May for A. S. M. E. Transactions, November 30, 1931.

"Oil Requirements for Modern Rolling Mill Circulating Systems," by C. M. Larson for A. S. M. E. Transactions, November 30, 1931.

"Lead Oleate Lubricants for Heavily Loaded Gears and Bearings," by Maurice Reswick, Iron Age, September 24, 1931.

80° F. Several times in the earlier operation we have had all of the oil in the entire system held up in the cases so that none would be left for pump feed to bearings, necessitating shutting down to drain the oil back. During the past year, with frequent shut downs, we have always cut back on the gear sprays to avoid this trouble until the mill was warmed up.

On a 4500 H.P. reduction gear set driving a 160" Sheared Plate Mill, it was decided to change from heavy gear shield or pinion tar to an oil circulating system while mill was being relocated. The clearance between the main gear and the gear case was so small, that when oil was sprayed between pinion and gear, the pumping action of the gear carried the oil all the way around the gear, and forced it out between the pinion shaft and gear case. In order to overcome this, a deflector baffle and additional drain was installed as shown in Figure 1. The peripheral speed of this gear is 2802 ft. per minute. When this gear set was relocated, the main pinion bearings being worn were rebabbitted, thus slightly changing the centers from those to which the pinion and gear had worn themselves. The result was that the tooth contact was changed slightly and when they were started up on oil, pitting was noticed and this continued until the teeth had worn in to sufficient bearing to reduce the concentrated load on a few points of the teeth below the stress failure point of the metal. As soon as this happened, pitting stopped and the teeth began to take on a high polish. This phenomenon has been ob-

GEAR AND PINION MEASUREMENTS—27", 24" AND 21" MILLS

27" MILL

			9-2-28	4-14-29	9-15-29	1-31-32	Total Wear
Mill Pinion #1	Driver	Top	1.400"	1.400"	1.400"	1.395"	.005
Mill Pinion #1		Bottom	1.419	1.419	1.420	1.404	.015
Gear #1			1.428	1.428	1.428	1.423	.005
Pinion #2		Top	1.424	1.423	1.418	1.385	.039
Pinion #2	Driver	Bottom	1.408	1.408	1.392	1.390	.018
Gear #2			1.431	1.428	1.431	1.418	.013

24" MILL

			9-2-28	4-14-29	9-15-29	1-31-32	Total Wear
Pinion #3	Driver	Top	1.395	1.395	1.393	1.390	.005
Pinion #3		Bottom	1.430	1.424	1.425	1.410	.020
Gear #3			1.439	1.430	1.428	1.424	.015
Intermed. Gear			1.431	1.417	1.428	1.420	.011
Motor Pinion			1.439	1.420	1.410	1.408	.031
Pinion #5	Driver	Top	1.377	1.377	1.375	1.386	+.009
Pinion #5		Bottom	1.417	1.415	1.425	1.415	.002
Gear #5			1.425	1.423	1.423	1.419	.006

21" MILL

			9-2-28	4-14-29	9-15-29	1-31-32	Total Wear
Pinion #6		Top	1.418	1.415	1.414	1.402	.016
Pinion #6	Driver	Bottom	1.390	1.390	1.385	1.368	.022
Gear #6			1.432	1.428	1.425	1.425	.007
Pinion #7	Driver	Top	1.376	1.375	1.373	1.360	.016
Pinion #7		Bottom	1.420	1.413	1.414	1.395	.025
Gear #7			1.425	1.425	1.424	1.421	.004
Intermed. Gear			1.435	1.425	1.425	1.425	.010
Motor Pinion			1.425	1.416	1.420	1.405	.020
Pinion #9	Driver	Top	1.412	1.413	1.405	1.395	.017
Pinion #9		Bottom	1.413	1.409	1.416	1.412	.001
Gear #9			1.412	1.413	1.413	1.413	.001
Pinion #10		Top	1.400	1.397	1.398	1.390	.010
Pinion #10	Driver	Bottom	1.440	1.370	1.372	1.370	.070
Gear #10			1.415	1.413	1.413	1.413	.002
Pinion #12		Top	1.425	1.422	1.416	1.414	.011
Pinion #12	Driver	Bottom	1.415	1.405	1.397	1.395	.020
Gear #12			1.400	1.400	1.400	1.413	....
Pinion #13	Driver	Top	1.417	1.402	1.395	1.370	.047
Pinion #13		Bottom	1.400	1.404	1.404	1.415	+.015
Gear #13			1.424	1.420	1.417	1.410	....
Pinion #14		Top	1.405	1.404	1.405	1.400	.005
Pinion #14	Driver	Bottom	1.419	1.414	1.412	1.410	.019
Gear #14			1.410	1.425	1.425	1.416	....
Motor Pinion			1.405	1.405	1.400	1.384	.021

* Gears 12, 13 and 14 are new due to change in gear ratio.

served in many cases in starting new units whether they be lubricated with gear shield or with oil. The initial pitting frequently causes alarm and the resort to heavier oils. Soon the teeth wear to a bearing and the heavier oil is given the credit for stopping the pitting, whereas in reality it had little if anything to do with it. It would seem that the users of continuous oiling systems should strenuously combat the growing tendency to recommend higher and higher viscosity oils for heavy mill drives and should insist instead on a cure for their apparent need. Before we accept these heavier lubricants we should require that our reduction gear sets and mill pinions have tooth forms that will give as high a percentage of true rolling action as possible, that these teeth shall be cut on machines that are in the best of condition so that accuracy of tooth form is maintained, that these teeth shall be filed into a bearing during assembly (in case where tooth hardness will permit) and that gear and pinion material shall be of the best. That tooth form and hardness is in many cases more of a factor than the viscosity of the lubricant is shown by the case of an 18" 6 stand continuous billet and sheet bar mill where oil of the following specifications was used on the mill pinions for 15 years:

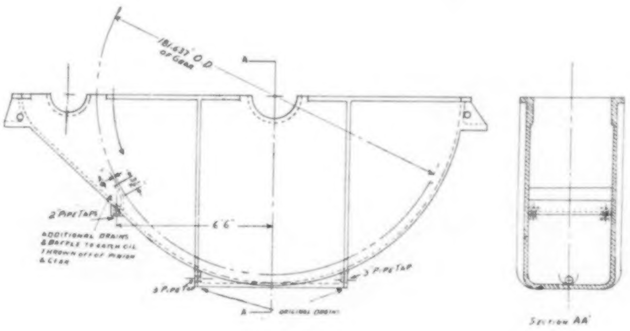

FIG. 1.

Gravity	25° Be
Cold Test	30° F.
Flash	390° F.
Viscosity at 100° F.	210 secs.
Viscosity at 210° F.	46 secs.

For the past year a slightly heavier oil of 285 secs. viscosity has been used but for a reason other than that of having a higher viscosity. On this mill on one stand with certain weights of bar, the tooth pressure per inch of face reaches 7950 lbs. and for average conditions is 6045 lbs.

Frequently more damage is done to teeth in large mill drives in the first few weeks of operation than in an equal number of years of subsequent operation. There does not seem to be any logical explanation of this except that when first started, the teeth did not have a uniform bearing from one end to the other, or had interference in entering or leaving, or were irregularly spaced, or had varying hardnesses. In other words, the actual mechanical form and alignment and composition of gear and pinion teeth for rolling mill drives is a most important and sometimes underrated factor, which should be given more consideration in order to render unnecessary the more expensive high viscosity oils that are more difficult to handle and clean.

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Micarta Roll Neck Bearings

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By W. B. SHIRK
Steel Mill Engineer, Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa.

To be presented before A. I. & S. E. E. Annual Convention in Pittsburgh, June 20-23, 1932.

Micarta is the trade name for Westinghouse products manufactured by impregnating sheets of fibrous material with a Phenol-Formaldehyde resin compound and then super-imposing one sheet upon the other in order to obtain a laminated mass of the desired thickness. The mass is baked under heat and pressure which converts the fibrous material and impregnating compound into a dense tough substance which has physical properties as shown in Figure I.

W. B. SHIRK

The suitability of Micarta as a roll neck bearing material became apparent several years ago when a mill superintendent became eager to find a bearing material which had a longer life and lower coefficient of friction than the conventional type of brass bearings, built and installed several bearings on a continuous mill. The performance of the bearings was so gratifying that additional mill stands were immediately equipped with Micarta bearings.

Micarta bearings can be obtained with the lami-

FIGURE I

PHYSICAL PROPERTIES OF MICARTA ROLL NECK BEARING MATERIAL

Compression—Edgewise	24,000 lbs. per sq. in.
Compression Modulus—Edgewise	400,000 to 500,000
Compression—Flatwise	44,000 lbs. per sq. in.
Compression Modulus—Flatwise	400,000 to 500,000
Brinell Hardness—Edgewise	32 to 40
Specific Gravity	1.38
Water Absorption	6% maximum saturation
Oil or Grease	Slight absorption
Dilute Acids—Very Good Resistance	Slight absorption
Dilute Alkalies—Fair Resistance	Slight absorption

nations parallel to the roll necks as shown in Figs. 2 and 3. However, from the very beginning of the application of Micarta to roll necks it was recognized that better bearing life would be obtained if the laminations were perpendicular to the roll neck as shown in Figs. 5 and 6. The increase in bearing life is due to the same factors which have made Micarta gears successful, wherein the laminations are perpendicular to the tooth load. With the laminations perpendicular to the load, constant wearing qualities are maintained throughout the life of the bearing.

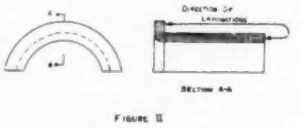

FIG. 2—Sketch of a Micarta Bearing Using Laminations Parallel to Shaft and Wound in Tube Form.

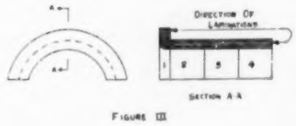

FIG. 3—Sketch of a Micarta Bearing Using Laminations Parallel to the Shaft. The Radial Portion of the Bearing Being in Sections.

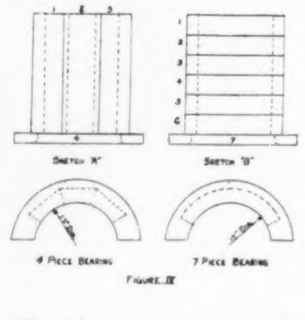

FIG. 4.
Sketch "A"—Radial Bearing Made from Strips of Micarta.

Sketch "B"—Same Size Radial Bearing Made from Section of Tube.

The supporting structure for Micarta bearings when applied to existing mill stands is influenced by the modifications that are necessary in order to use wherever possible the existing bearing shells and chucks. A mounting which has given very satisfactory results is shown in Figs. 7 and 8. The detachable thrust block is held in place by wooden or Micarta dowel pins which fit into the bearing shell. In addition to the dowel pins the face of the shell is recessed so as to prevent the movement of the thrust block around the shaft. To prevent the radial bearing segments from moving in the shell two steel strips running parallel to the axis of the shaft are bolted, screwed or welded to the shell which firmly holds the radial bearing. To prevent the radial bearing moving parallel to the shaft when thrust occurs two wooden or Micarta dowel pins are inserted in the end radial bearings segment. If dowel pins are not desired a steel strip placed across the face of the shell and below the bearing surface will hold the radial bearing satisfactorily.

Another method of mounting the bearing is shown in Fig. 9. This method is quite similar to the method shown in Figs. 7 and 8 with the exception that the two metal strips not only hold the radial bearing in place but they also hold the shell in the bearing chuck.

A Micarta bearing like all other types of open bearings should be provided with lubrication grooves and in addition the bearing should be relieved on either side in order to insure adequate lubrication and cooling for the load carrying portions. Field tests have proven that a grooved bearing gives longer life than a plain bearing and by providing the grooved bearing with an adequate relief on both sides, the life of the bearing will be further increased over the plain bearing.

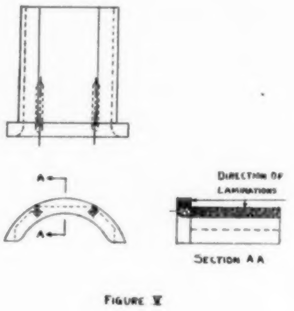

FIG. 5—Sketch of Micarta Bearing with Machined Back to Fit Bearing Shell.

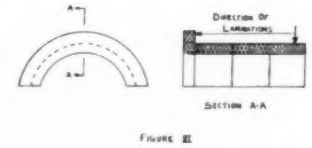

FIG. 6—Sketch of Latest Type of Micarta Bearing.

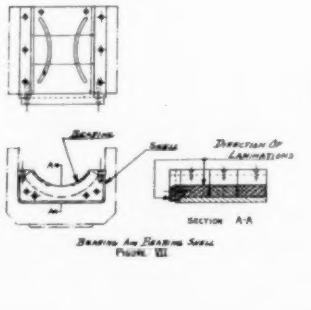

FIG. 7—Sketch Showing Suggested Mounting of a Micarta Bearing in a Shell.

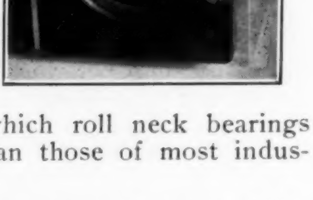

FIG. 8—Photograph of a Bearing and Shell Similar to Figure 7.

The conditions under which roll neck bearings operate are more severe than those of most industrial bearing applications. Roll neck bearings of the open type are traditionally heavily loaded and poorly lubricated. A roll neck bearing is also subjected repeatedly to severe impact or shock loads every time the steel enters the pass. The constant presence of mill dust and scale constitute additional detrimental circumstances. The rapid wear of roll neck bearings is also objectionable because, off-gage material will result and therefore frequent objectionable bearing changes are necessary.

The first experience one obtains after installing any type of bearing is the static friction encountered when the mill is started. The starting torque required to start the mill is a function of the static coefficient friction and the static load on the bearings. With the roll necks stationary and before a complete revolution of the neck is made it is reasonable to assume that no lubricating film exists between the bearing and the roll neck. Under such conditions we can assume that practically a dry bearing exists.

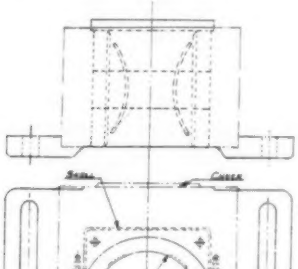

FIG. 9—Sketch Showing Suggested Method of Holding Bearing and Shell in Chuck.

In order to obtain an idea of the comparative value of static friction when using Micarta bearings and the conventional type of brass bearings under different conditions, extensive field tests were made. The curves shown on Fig. 10 were not taken in order to establish the coefficient of static friction but were taken to merely determine the comparative characteristics for the bearings under certain conditions.

Each bearing for the test shown in Fig. 10 was run in and then thoroughly cleaned to remove any foreign material. The bearing was then replaced in the test rig and allowed to run for approximately five minutes in order to seat itself properly and then shut down. Each bearing under the conditions shown on the curve was then started at intervals of 5, 10, 15, 30, 60, 120 and 180 minutes. From the results obtained it will be noted that the static friction for dry brass bearings is higher than the static friction for dry Micarta bearing. During the test, water was flushed on the journals and the term "dry" is used to point out that the bearing surface in contact with the journal was assumed to be dry.

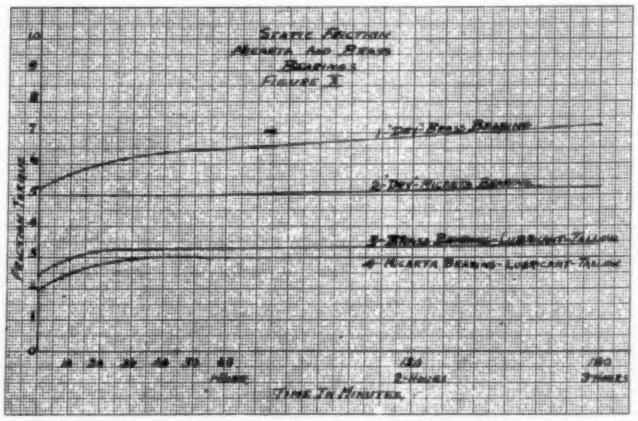

FIG. 10—Static Friction Curves Showing Comparative Results of Field Tests on Micarta and Brass Bearings.

Occasionally it is stated that after the mill is set up and the screwdowns properly adjusted it is difficult to start the mill when using Micarta bearings. Since Micarta bearings give longer life than brass bearings under similar conditions we attribute the increased starting load to the fact that decreased rate of wear results in small bearing clearances and therefore the starting load is slightly increased when compared to brass bearings.

Micarta bearings lubricated with tallow have less static friction than brass bearings under the same conditions. Low journal speeds require high viscosity lubricants therefore since tallow has a very high viscosity and not readily pressed out of the bearing it is only natural that the static friction of bearings lubricated with tallow will be less than "dry" bearings under similar static load conditions.

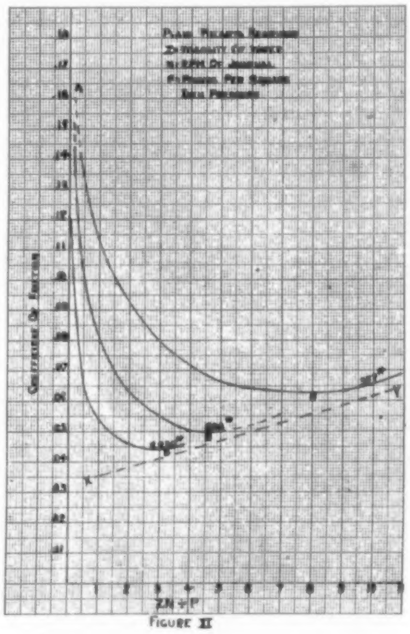

FIG. 11—Laboratory Test Curves on Small Plain Micarta Bearing at Various Loadings.

If conditions arise where an increased starting load is encountered the mill may be satisfactorily started by backing-up on the mill stand screwdowns or by backing-up on the mill and then starting by taking up the slack in the mill.

In order to determine the load carrying capacity of Micarta bearings a series of laboratory tests were made on small bearings lubricated and cooled with water only. The results of the tests are shown on Fig. 11 for plain bearings and Fig. 12 for graphitized bearings. It will be immediately observed that graphitized Micarta bearings have a lower coefficient of friction for a given speed and pressure than plain Micarta. The lower coefficient of friction is attributed to the lubricating qualities of the graphite which is distributed uniformly throughout the bearing structure.

From the curves in Figs. 11 and 12 it will be noted that an interesting bearing phenomena takes place. When the loading remains constant the coefficient of friction decreases to a point "B" as the journal speed is decreased. After passing the point "B" to the left, we lose the lubricating film and the coefficient of friction increases rapidly for a small decrease in speed. We attribute the increase in friction to the fact that a solid and a liquid film exist. On the other hand if the speed is kept constant it will be observed that the coefficient of friction decreases with increased loading.

The portion of the curves extending from point "B", up toward the right, indicates that a perfect liquid film exists. Since the curves to the right of point "B" appear to be straight lines when extended, the same bearing laws that apply to sleeve bearings now apply to Micarta bearings which states that when a perfect film exists between journal and bearing the pressure multiplied by the coefficient of friction is equal to a constant when the speed is constant.

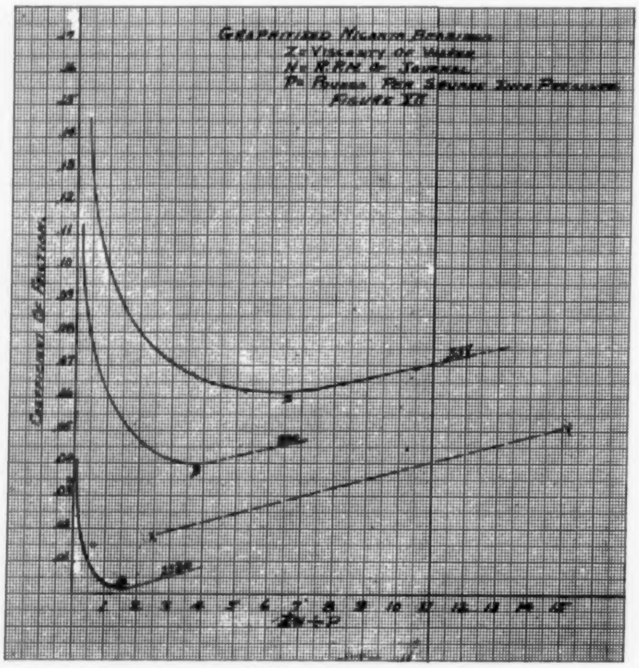

FIG. 12—Laboratory Test Curves on Small Graphitized Micarta Bearings at Various Loadings.

The theory as regards the excellent bearing qualities of Micarta and water lubrication have been verified by actual field experience. Records have been kept on mills using Micarta bearings which show that the life of the Micarta bearing is from 10 to 15 times the life of conventional brass bearings. The kilowatt hour per ton energy consumption has been reduced from 10% to 25% when compared with the conventional type of brass bearing. Due to the reduced coefficient of friction the rolling peak load has been reduced as much as 25% when compared with brass bearing. Fig. 13 shows a comparative field test on Micarta bearings lubricated with water and brass bearings lubricated with a combination of grease, suet and water.

Some operators have attempted to determine the life of the Micarta bearings as compared with brass bearings on the basis of how the Micarta will stand up when rolling the heaviest schedule on the mill. This is not a fair trial for any type of bearing because the life of a brass bearing is the result of rolling a wide variety of schedules on the mill from the lightest to the heaviest schedules included. Observations on mills 16 inches and smaller have indicated that the majority of roll necks are very rough and scored by the use of brass bearings. Scored necks prove that no lubricating film exists and therefore with metal to metal contact the roll neck being harder than the brass will naturally cause rapid wear. With Micarta, the neck is not scored due to the absence of a lubricating film but is polished to a very high degree which in turn results in lower friction losses and longer bearing life.

Numerous installations of Micarta bearings are carrying loads ranging from 350 pounds per square inch of projected area at 1800 feet per minute to 4000 pounds per square inch at 450 feet per minute on the journal when lubricated with water only. Field experience has proven that increased life of the Micarta bearing is obtained if water only is used as a lubricant. In order to obtain the maximum bearing life for any type of bearing the acidity of the cooling water should be reduced to less than 15 parts in a million. The acid in the water does not attack the Micarta but has a tendency to roughen the roll neck.

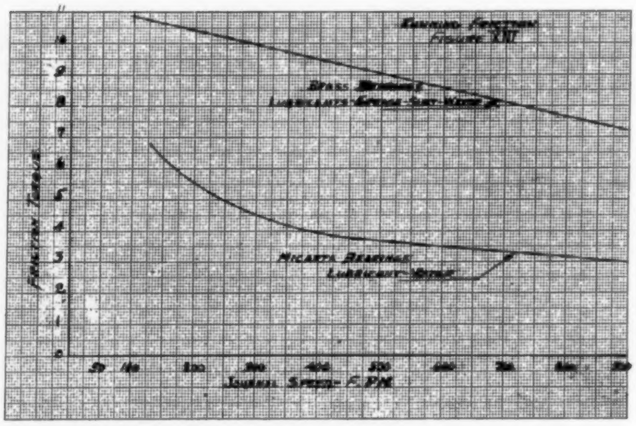

FIG. 13—Running Friction Curves Showing Comparative Results of Field Tests on Micarta and Brass Bearings.

One or two instances have occurred where the mill operators have stated that they were not able to hold gage on the finished product when using Micarta bearings. The question of holding gage on a mill is similar in some respects to the problem of holding a loop between adjacent stands of a tandem mill. If you set the no load speed correctly so that the desired rolling speed is obtained a satisfactory loop will result. The same is true when using Micarta bearings. If the mill screws are set so as to take into account, the spring in the rolls, stretch in the mill housing and the resilient characteristics of the Micarta bearings a finished gage within commercial tolerances should be obtained.

There are additional advantages to the use of Micarta bearings that a careful analysis of the true fact will bring out. Since it is recognized that Micarta bearings using water as a lubricant have a longer life than brass bearings under comparable conditions we immediately eliminate the oil, grease or suet required for brass bearings which is a considerable saving. Longer life of Micarta bearings means less rate of wear and therefore less screw-down adjustments which results in a more uniform finished product. The renewable Micarta thrust block has the advantage that if rapid wear takes place on the thrust block, the radial bearing is not lost and renewal of the thrust block is all that is required. Due to the fact that Micarta is resilient more consistent roll alignment is maintained by the use of Micarta thrust blocks. Experience has proven that in the case of brass bearings when the mill is reversed or backed-up you lose your roll alignment. The life of the Micarta thrust block is also many times the life of the brass thrust face for the same reasons that increase the life of the radial Micarta bearing.

Since Micarta is an insulator the cooling of the bearing is effected entirely by the amount of water flushed on the roll neck. Observations in the field show that the bearings operate at very low temperatures because the human hand can be kept continuously on the bearing while in operation, without any uncomfortable results. In a mill using Micarta bearings the main drive motor should be shut down immediately in case the water supply fails otherwise the bearings may be damaged. Unlike brass bearings in case of water failure the Micarta bearing will not roughen or destroy the roll neck surface. Micarta bearings like all other type of bearings should only be used on new or resurfaced roll necks otherwise the life of the bearings will be considerably reduced.

Micarta bearings unlike other types of open bearings give excellent life under adverse conditions where poor lubrication and good cooling conditions prevail. With the successful introduction of the roller bearings for large mill stands supplemented by the use of Micarta bearings, mill operators have new equipment available in order to effect additional economies in the rolling of ferrous and non-ferrous metal, hot or cold.

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Progress in The Electric Welding Science

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To be presented before A. I. & S. E. E. Annual Convention in Pittsburgh, June 20-23, 1932.

In reviewing recent progress in the science of electric arc welding, first place today must be given to the improvements in quality of weld and speed of operation which accompany the use of heavily coated electrodes for the welding of steel parts. The ductility of arc deposited steel, (by which we really mean the **plasticity** of the metal or its ability to deform permanently without fracture) as measured in the tension test, has been increased 100 per cent or more, that is from about 5 to 10 per cent in a two-inch length up to 20 to 30 per cent. Such plasticity in a casting, and particularly in one made as in arc welding from super-heated steel, with little or no chance to stand while liquid and thus separate the entrapped impurities, and one "cast", moreover, in a chill mould, is a remarkable achievement. While these results were achieved practically as much as 5 or 10 years ago by one company, the A. O. Smith Co., it is only within the last year or so that electrodes of this quality have become available commercially.

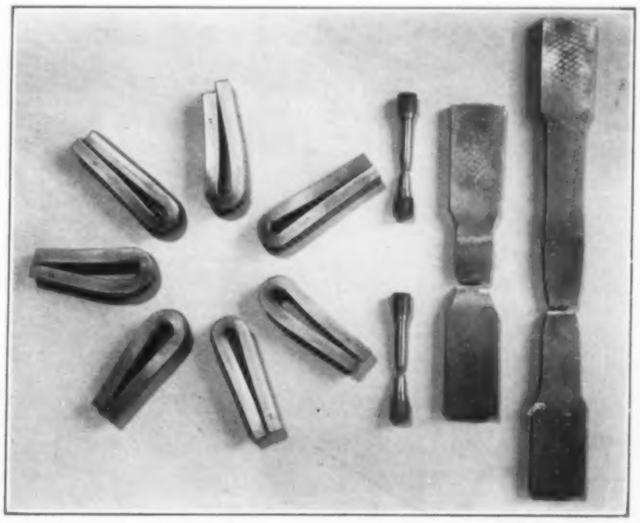


FIG. 1—Ductile Arc Weld Metal. Tension Specimens are All-Weld Metal.

The property of plasticity in a weld is important for several reasons, first of which is the relief of the shrinkage stresses which are set up during cooling of the weld and adjacent parts. Such stresses in a weld often result in cracking of the weld during cooling, if the weld metal is not plastic. If cracking does not develop at this time, then later when an external stress is added to the internal stress in the weld, fracture may occur and constitute a failure of the weld in service. Sometimes the seasonal change in temperature or the daily changes alone will be sufficient, when added to the internal stress, to cause failure if the weld is brittle. When the weld metal is plastic, it yields to these shrinkage stresses by plastic deformation and the stresses drop to the yield point of the metal or below. Service stresses, if high, may later cause a continuation of this plastic deformation without fracture of the weld, whereas a brittle weld would fracture.

The procedure by which higher ductility is obtained is to exclude as far as possible the reactive oxygen and nitrogen of the air from contact with the weld metal, both at the tip of the electrode and in the crater. Oxides and nitrides of iron, which form if these gases are present in welding, are soluble in the grains of iron when rapidly cooled but later, upon standing at room temperature, these dissolved impurities begin to precipitate and render the weld hard and brittle. The process is similar to the ageing of duralumium. Its role in welding is the discovery of Messrs. Hensel and Larsen of the Westinghouse Company. The coatings used on the electrodes are either cellulose or fusible mineral substances and these envelope the arc and the crater and diminish the amount of oxide and nitride entering the weld, so that the hardening and embrittlement do not take place upon cooling. The result is a ductile weld.

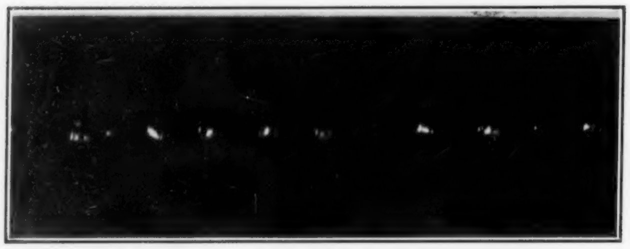


FIG. 2—Separated Deposits During Arc Welding.

Even the purer and more plastic metal so deposited would still be somewhat weak due to its ingot structure resulting from casting. If a second layer of weld is deposited on top of the first layer, there is some re-crystallization and partial destruction of the ingot structure and consequent improvement in properties of the first layer due to the re-heating. If the first layer is hammered (peened) re-crystallization will be more extensive. Re-heating also removes residual stresses in the first layer, and by re-melting its top surface removes some included slag particles and porosity. The net effect on the previous layer is to refine its grain structure, free it of internal stress and partially of porosity and inclusions.

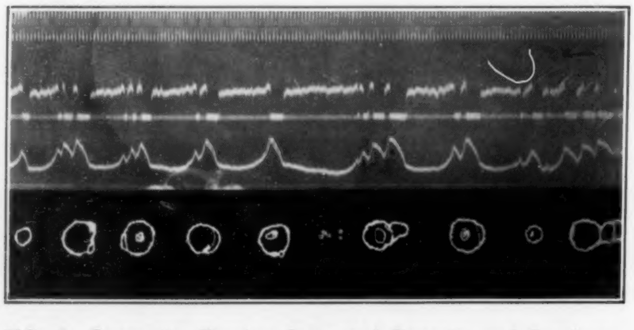


FIG. 3—Composite Showing Separated Deposits and Synchronized Oscillogram.

It is this quality of weld which has led the A. S. M. E. Boiler Code Committee to accept fired-pressure vessels of welded construction and the Babcock and Wilcox Co. and others to manufacture them. Such acceptance will no doubt soon be extended to other structures where risk is involved.

One of the remarkable accompaniments of the process is an increase in the speed with which welding is done. Whereas the bare electrode deposits metal at the rate of perhaps 5 to 10 pounds of metal per hour, the heavily coated electrode may be used to deposit 25 to 30 pounds in the same time.

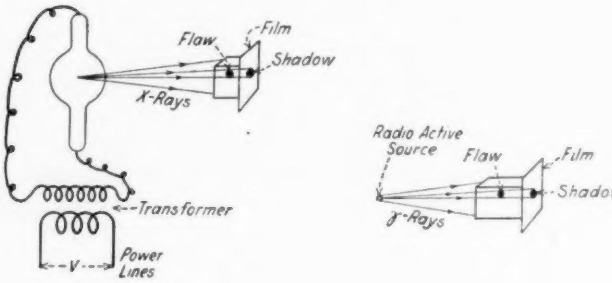
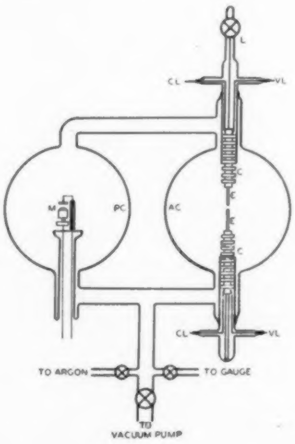


FIG. 4—Radiography by X-Rays and by Gamma Rays.

To electrical engineers the electrical phenomena of the arc during welding are of interest. High speed motion pictures have shown that the metal traverses the arc sometimes as globules and sometimes as threads. An oscillograph, synchronized with the camera, shows that in "short-arc" welding the arc is short-circuited with each transfer of metal. This happens several times per second. At these times of short-circuit the voltage approaches zero and a high transient current flows. More recently we have devised a means for recording the deposits by moving a polished plate rapidly under the arc so that the deposits appear separated **in space** on the plate as they were separated **in time** in the arc. An oscillographic record correlated with a deposit of this kind is shown in Fig. 3. The correspondence

FIG. 5—Pure Iron Arc in Argon.



between the two records is surprisingly good, showing that the oscillograph is a good indicator of the exact mode of metal deposition by the arc at any instant. These studies are of interest to the metallurgist in improving the flowing properties of the electrode and also to the electrical engineer in the design of a machine to accommodate transients.

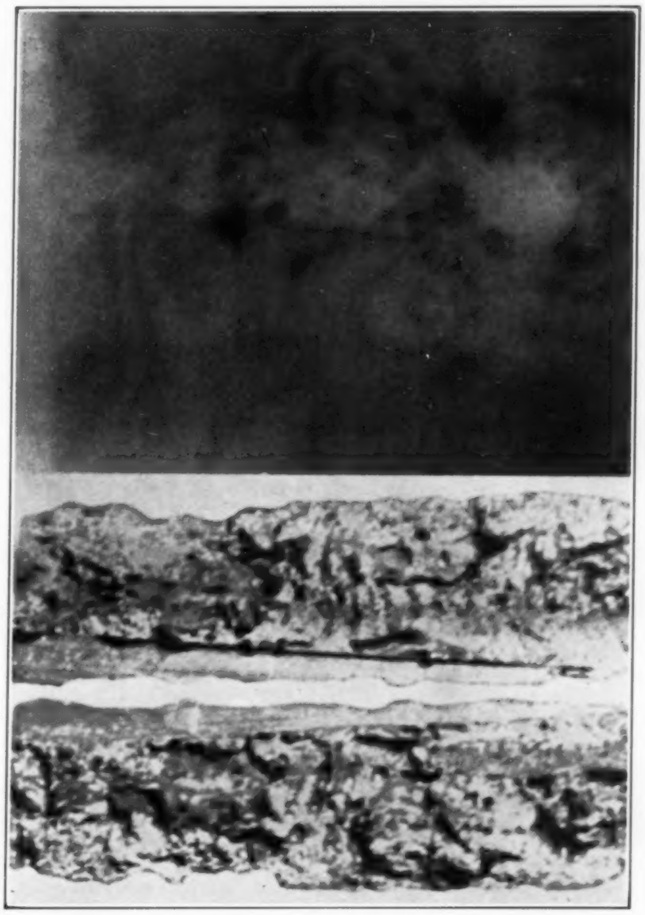


FIG. 6—Radiograph and Fracture of Porous Weld by Gamma Rays.

One is struck, in studies of arc welding, with the perplexing number of variables always involved in the operation: composition of the steel being welded and composition of the electrodes; current, voltage and arc length, electrical characteristics of the machine, cooling rates, etc. When especially good or bad results are obtained, it is impossible to attribute them to any one factor. Obviously, thousands of dollars have been wasted in this maze. Recently the A. I. E. E. in an attempt to establish some basis of reference requested Engineering Foundation to sponsor some studies using **pure iron,** so that we could know the results which pure iron would yield in welding, and then later find the true role of oxygen, carbon, phosphorus, etc., in the steel to be welded and in the electrodes.

Spectrascopically pure iron was contributed by the General Electric Co. and an arc was struck in very pure argon gas at approximately atmospheric pressure. I say an arc was struck, but actually I should say that it was found impossible to strike an arc even with 120 volts. If the purification of the argon were slightly less drastic than that which we used, the normal arc was obtained. For example, in argon of total impurity of .0000003 per cent, the normal arc appeared, but in the further purified gas only a weak glow, going out upon separation of the electrodes, set in. Evidently, although not recognized previously, an arc does depend on the presence of chemically active atoms or molecules. These studies are being continued, one student having gone as Research Fellow from Lehigh to Berlin to continue the studies under Hertz, and two more having undertaken the continuation of the work at Lehigh.

Not only the electrical characteristics of the arc but also the metallurgical properties of the pure iron deposited under these pure conditions, and later in air, are being studied. These metallurgical studies will be ready for report shortly.

Finally, although I must be conscious that this choice will exclude many other developments worthy of a place, I may mention the portable method for radiographic testing of welds and other castings, which employs the gamma rays from radium or its derivatives. Although the softer, longer X-rays are inherently better suited for examining thin materials for minute flaws, and give better detail, the cumbersome transformer, control devices and tube render it well nigh immovable from the laboratory. A bulb of radium, or its gas "emanation", the size of a pea on the contrary may readily be carried to the top of a building, along a pipe line or into the hold of a ship, and its sensitivity permits it to reveal flaws as small as 2 per cent of the total thickness of the weld or casting examined. Its sensitivity is therefore adequate for the examination of welds.

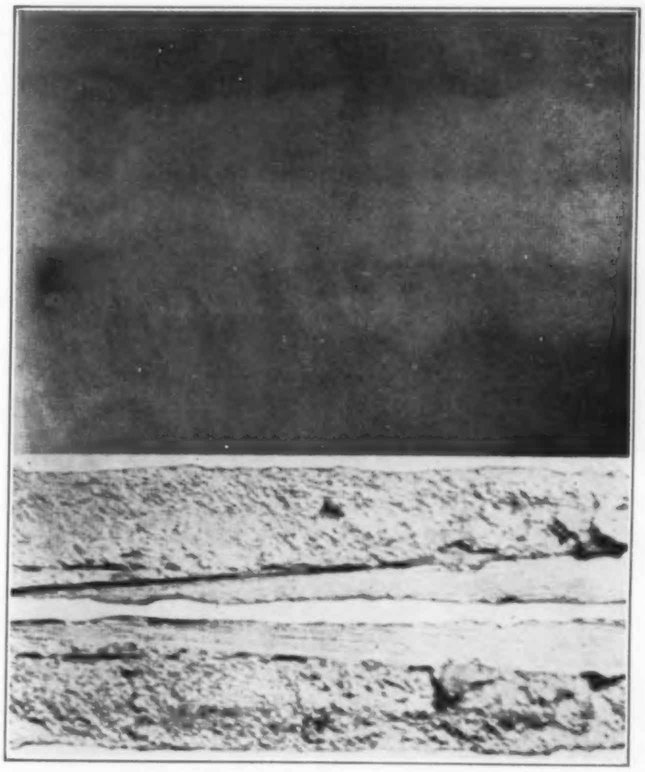


FIG. 7—Radiograph and Fracture of Sound Weld by Gamma Rays.

Figs. 6 and 7 illustrate the results obtained in several specific instances. There are, no doubt, many cases in which it will be useful.

Studies in other fields are not being neglected. The endurance life of welds under alternating stresses is being studied. The question of orientation of grains in a weld is yielding to the X-ray method. Creep phenomena and many other phases are being explored both under the guidance of the American Bureau of Welding and under other auspices. While many of the advances made in a technical process come from invention and from the Edison method of cut-and-try, the persistent analytical study of the welding process by the methods of scientific research seem also to yield valuable results.

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Determination of Motor Capacity For Main Roll Drives

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To be presented before A. I. & S. E. E. Annual Convention in Pittsburgh, June 20-23, 1932.

In selecting the motor drive for a rolling mill, it is, quite naturally, essential that full details as to the mill itself be available. These include mill layout, roll diameters, roll speeds, and gear ratios if these have already been selected. It is also essential that complete proposed rolling schedules, or at least those which will impose the heaviest work, and demand the highest and lowest mill speeds, be at hand. These should include: area, shape, length, weight, and temperature of entering ingot, slab, bloom or billet; number of passes; shape and area after each pass; roll speed or delivery speed of each pass, interval between passes; maximum number of pieces rolled per minute; possible overlapping of various passes; any special considerations.

H. A. WINNE

Having this information, there are several ways to determine what the capacities of the motors must be. Various formulae have been published from time to time, but all I have seen involve some "constant" (really quite variable) or constants which must be determined from test data, and a great deal of judgment and past experience is necessary in their use. Very frequently too the calculations involved are long and laborious, too much so to make them feasible for use where an engineer must determine the power requirement of a good many rolling schedules in the course of a day. Descriptions of some of these methods are given in papers listed in the bibliography following this article.

The method which has been used, I think I am safe in saying, in selecting by far the majority of the electric main roll drives now installed in this country is very briefly as follows: First the amount of deformation of the metal in each pass, expressed in elongations, volume displaced, or other terms, is determined. Then from test data on mills of the same general character, plotted in such form that the work per unit deformation is readily obtainable, the power required for each pass is calculated. This general method, in one form or another, has been largely used by the electrical manufacturers. It is the purpose of this article to describe in detail this method, and to give a sufficient amount of data so that drives for the more common types of mills may be selected.

It is evident that the use of such a system presupposes the existence of a considerable amount of test data. In the early days of motor application such data was meagre, and was of necessity obtained from steam driven mills. If one has attempted to secure accurate test data on a steam driven mill, and to interpret the results, he will appreciate the difficulties of such an undertaking. Nevertheless, it can be done with a fair degree of accuracy. One of the earliest attempts to collect a large amount of power data on steam driven mills was by a German engineer, Dr. J. Puppe. His work is almost a classic, and formed the basis for calculation of many of the earlier motor drives.

One of the advantages of motor drive is the simplicity of measuring the power input, and as the number of motor driven mills increased, power data was rapidly collected which soon superseded most of the older data on steam driven mills. This collection of data is still going on, and it is one of the purposes of this article to outline the method of taking power tests on mills and putting the data in usable shape, with the hope that steel mill operating engineers will be encouraged to collect such information and publish it for the good of the industry at large.

MAKING POWER TESTS

Since data must be obtained before it can be used, it seems logical first to discuss the method of making tests on existing mills.

The information we are after can be divided into two basic classes: First, the amount and rate of deformation or elongation of the metal in each pass; second, the power delivered to the roll stand. The first may be called the rolling data, the second the power data. Of the two the rolling data is usually the more difficult to get.

Obtaining the Rolling Data

First we should have the shape, area, length, weight, and preferably temperature, of the piece of steel entering the mill. These are all fairly easy to obtain, with the possible exception of the temperature, and if radiation or optical pyrometers are available this difficulty disappears.

Reduction Per Pass

Next, in order to get the deformation in each pass we need the area of the metal leaving each pass. This is hard to secure accurately, but several methods give reasonably close results. Possibly the most accurate is to shear off a piece of the bar after each pass, weigh a measured length, and calculate the area, figuring the weight of hot steel at about 0.275 pounds per cubic inch. The mills on which this is feasible are few and far between.

Possibly the simplest method is to take the roll designer's drawings, or templates, and planimeter the area. This, of course, makes no allowance for roll spring, or incorrect roll setting. This method cannot be used on a mill on which the roll spacings are adjusted for different schedules, as for example a blooming mill or continuous hot strip mill.

If it is possible to measure the length after each pass, comparison of this with the original length and area gives an accurate measure of the area, since

$$Lo \times Ao = Ln \times An \quad \text{.......} (1)$$

where Lo and Ao are the original length and area, and Ln and An are the length and area after pass n.

In testing a blooming mill we usually must depend on indications of the screwdown gauge, estimating the amount of side spread.

On a hot strip mill we may get an opportunity to gauge the thickness after each pass, or we may have to depend on the settings of the screwdowns. It is always possible to gauge the thickness of the finished strip, and from this and the delivery speed of each stand, we can calculate to a fair degree of accuracy the thickness of the strip leaving each **continuous** stand. For example, supposing we are delivering strip 0.09″ thick at 1000 FPM from the last stand. Then if the next to the last stand runs at 900 FPM, and the metal is continuous between the two, with inappreciable looping or stretching, the thickness out of this stand must be $\frac{0.09 \times 1000}{900} = 0.1''$. This is evident when we remember that if the metal is continuous from stand to stand, without looping or stretching, the same **weight** of metal must pass through each stand in the same length of time. In other words, for continuous stands

$$A \times S = K \quad \text{.......} (2)$$

where A is the area, S the lineal speed, and K a constant.

In all tests care must be exercised to insure that the very fact that measurements are being taken does not lead to incorrect results. For instance, if the length after each pass is being measured, the operation of the mill may be delayed enough so that the steel will be colder than normal in the latter passes, resulting perhaps in excessive spring of the mill, and certainly in more than normal power consumption. In order to get accurate results it may be necessary to measure a few passes on one piece, a few others on the next, and so on, then taking all power readings on pieces on which no length measurements are attempted. This, of course, presupposes that all entering pieces are of the same size, and uniformly heated.

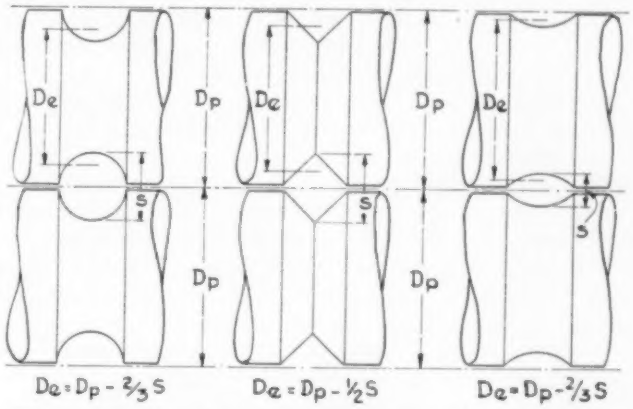


Fig. 1 Effective Roll Diameters (De) for Different passes

Delivery Speed

Having determined the area, the next consideration is the delivery speed. While it is not necessarily exactly accurate to say that this is equal to the peripheral speed at the **effective** diameter of the rolls, we find that this answers all practical purposes. There may be some "slip", or in some cases a "squirt", or overspeeding of the metal, but the percentage is small.

The r.p.m. of the rolls is usually easy to determine, but the effective or working diameter on such a mill as a structural or rail mill can only be estimated. On strip mills and blooming mills the effective and actual diameters are obviously identical. In Fig. 1 are illustrated diagrammatically some of the "rules of thumb" which are used in estimating the effective diameter for various pass shapes.

Obtaining the Power Data

To make a power test only very simple electrical instruments are really necessary. High speed graphic meters and similar refinements make the work somewhat easier, but in most cases add little if anything to the accuracy, and can be dispensed with. For a synchronous or induction driving motor, without flywheels, an indicating wattmeter will suffice. For a direct current motor an ammeter and a voltmeter will answer in most cases. Tachometers are essential on adjustable speed drives.

We want, of course, to obtain the power input for each pass independent of the others. If each stand is driven by a separate motor this is very simply done. If several stands are driven by one motor, we may have to take one reading with metal in pass one only, another with passes one and two filled, and so on. Then we can segregate the power for, say, number two pass by subtracting the first reading from the second. In such instances a graphic meter facilitates matters.

We find it advisable also to segregate the no load losses of mill and drive from the rolling work. To do this we read the input to the motor when there is no metal in the mill. Then if we subtract this reading from the input when one pass is filled, the result will be the sum of what we may term the "net rolling power" for that pass, plus the **"load losses"** in the motor. By "load losses" I mean the increment in the total motor losses which is caused by the load for any pass. These "load losses" are relatively a small percentage of the motor input and may be neglected without any great error.

The "running light" loss of the mill itself, segregated from the motor loss, is also desirable information for use in calculating the power requirements of other mills. To obtain this we must have some idea of the motor no-load losses.

Motor Losses

Where extreme accuracy is desirable the values of motor losses at various loads may be secured from the manufacturer. For most work the following rules, while by no means strictly accurate for all motors and loads, will give results sufficiently close.

For d-c. motors use as no-load loss, exclusive of excitation loss, at basic or full field speed, 1.3% of the motor rated horsepower. At speeds above basic, increase the loss in proportion to the speed. For example, a 1000 hp.-200/400 r.p.m. motor will have a running light loss at 200 r.p.m. of about 13 hp., not including shunt field loss. At 400 r.p.m. the loss will be in the neighborhood of 26 hp.

To obtain the "net rolling power" multiply the difference between the load input and the no-load input by 0.97.

For synchronous motors use as the no-load loss, exclusive of excitation, about 2% of the motor rating for a 0.8 PF motor, and about 1.7% for a unity power factor motor. The factor for "net rolling power" is about 0.99.

For induction motors without secondary resistor, the no-load loss is about 2.8% of the motor rating. The factor for "net rolling power" is about 0.98.

Drives With Flywheels

Obtaining power data on drives equipped with flywheels is quite a problem. The difficulty is to determine how much power is delivered to any pass by the wheel. If a high speed graphic wattmeter is available, and if the interval between passes is sufficient for the flywheel to return to full speed after each pass, we can integrate, by a planimeter, the area under the wattmeter curve from the beginning of the pass up to the point where the flywheel returns to full speed. This gives the hp.-seconds input to the motor during the pass. From no-load readings, and data obtained from the manufacturer, it is possible to calculate the hp.-seconds loss in the motor, secondary resistor and wheel during this time, and subtracting this from the total gives the "net rolling work" in hp.-seconds. Dividing this by the time metal is in the rolls in seconds, which can usually be determined from the record, gives the net total horsepower input to the mill during the pass. As a matter of fact, however, the hp.-seconds figure is just as usable as the horsepower, for we want to get our final result in terms of horsepower-hours per net ton of metal rolled.

If a high speed graphic wattmeter and a graphic tachometer can be had, with charts synchronized, the following method can be used. Select a point on the record for any pass and determine the slope of the speed curve and the horsepower input to the motor at this point. From the slope of the speed curve the rate of deceleration of the mill in r.p.m. per second can be determined. Then it is necessary that we find, either by calculation or from the manufacturers, the total WR^2, or inertia, of the motor, wheel, gears, and mill, all expressed in lb.-ft.^{2} referred to the motor shaft.

Then if a = rate of deceleration in r.p.m. per second

WR^2 = inertia in lb.-ft.^{2}

T_W = torque delivered by the wheels in lb.-ft.

$$T_W = \frac{WR^2 \times a}{307.6} \qquad (3)$$

The motor manufacturer can supply a curve showing the relation between h.p (orKW) input and torque output of the motor. From this curve the motor torque corresponding to the input at the selected point on the meter record can be read. Call this T_M.

The torque required to run the mill light can also be determined from the record and the curve. Call this T_F.

Then the net torque needed to turn the mill during the pass is $T = T_W + T_M - T_F$.

Then, if H = net rolling work in horsepower-hours per net ton

A = area of metal after pass in sq. in.

D = effective roll diameter in inches

$$H = \frac{T}{AD \times 136} \qquad (4)$$

The derivation of this formula will be given later.

Reversing Drives

A reversing drive is also difficult to test, for the motor armature has a considerable flywheel effect, and in normal operation it is accelerated and decelerated during the pass. The same general scheme as the last one described for testing a flywheel drive may be used, except that in order to determine the torque it is also necessary to know the motor field current. Usually an oscillograph is preferable to graphic meters for such a test.

A reversing mill may also be tested by having the operator run the passes through without accelerating or decelerating the drive during the pass. Then by reading the armature amperes and volts, the net rolling power may be calculated as previously described for a non-reversing drive.

Calculation of Results of Power Test

Probably the simplest way to show how the results of a power test are put into a form suitable for use in forecasting mill power requirements is to work through a concrete example.

A certain 18″ 5-stand continuous mill is rolling 8″ x 13.25 lb. sheet bar from 7¾″ x 2¾″ slabs. A high speed graphic wattmeter shows that the KW input to the 4000 hp. synchronous driving motor, which drives all stands through a lay shaft and bevel gears, is as follows:

TABLE I

With no metal in mill	225 KW input
With stand 1 filled	510 KW input
With stands 1 and 2 filled	950 KW input
With stands 1, 2, and 3 filled	1400 KW input
With stands 1, 2, 3, and 4 filled	1900 KW input
With stands 1, 2, 3, 4, and 5 filled	2400 KW input

Table II shows the tabulation of data and calculations for this test. Lines I to VI of the table are self explanatory. Unfortunately, temperature data was not taken, so line II is blank.

To obtain the motor input KW required for each pass, in line VII it is only necessary to subtract from the meter reading when that and all preceding passes are filled, the reading for the preceding passes. This KW value, of course, includes the additional losses in the motor, and therefore to get the net output of the motor for this pass we multiply by the factor 0.99 (since this is a synchronous motor), and obtain the values in line VIII. These are converted to horsepower by simply dividing by 0.746.

Now we have the horsepower for each pass, but we want to get the hp.-hrs. per ton of metal rolled through each pass. First we figure the tonnage which could pass through each pass per hour, assuming the pass to be continuously filled. Multiplying the area after each pass by the delivery speed **of that pass** in feet per minute, times 12 gives cubic inches per minute, times 60 gives cubic inches per hour, times 0.275 (weight of hot steel per cu. in.) gives pounds per hour, and dividing by 2000 gives net tons per hour. The results are given in line XI.

Now, if the mill ran full continuously, the horsepower input would remain continuously at the values given in IX. Consider pass 2. The net horsepower for this pass is 585. With the mill full, in one hour this pass would require 585 hp.-hrs. In this same hour 163 tons would be rolled. Therefore, this pass requires 585/163, or 3.58 hp.-hrs. per ton rolled. Going through the same calculations for each pass gives the results in line XII.

Line XIII gives for each pass the total hp.-hrs. per ton consumed up to and including that pass. It is simply the summation of the amounts in XII. Line XIV shows the total elongations made from the original slab to and including each pass. It is the ratio of the length after the pass to the original length, or the original area to the area after the pass.

To put the results of these calculations in convenient form we may plot the total hp.-hrs. per ton as ordinates, and total elongations as abscissae, re-

TABLE II

Power Test on 5-Stand Continuous Bar Mill—4000 HP. Synchronous Motor
Rolling 8″ x 13.25 lb. sheet bar from 7¾″ x 2¾″ x 30.5′ x 2150 lb. slabs, direct from bloomer.

Test Readings Pass Number	0	1	2	3	4	5
(I) Dimensions of metal	7¾″x2¾″	7.85″x2.19″	7.98″x1.56″	8″x1.12″	8″x.806″	8″x.59″
(II) Temperature	......	......	......	......	......	......
(III) KW Input, including all preceding passes filled	225	510	950	1400	1900	2400
Test Data						
(IV) Effective roll diameter	......	18″	18″	18″	18″	18″
(V) Roll RPM	......	20.0	27.6	38.3	53.3	72.8
(VI) Delivery speed FPM	......	94.5	130	181	252	344
Calculations						
(VII) KW Input per pass	225	285	440	450	500	500
(VIII) Net KW per pass	......	282	436	445	495	495
(IX) Net HP per pass	......	378	585	597	664	664
(X) Area of metal, sq. in.	21.3	17.2	12.4	8.96	6.45	4.72
(XI) Net tons per hour, 100% full	......	163	163	163	163	163
(XII) HP-Hrs. per ton per pass	......	2.32	3.58	3.66	4.07	4.07
(XIII) HP-Hrs. per ton, total	......	2.32	5.90	9.56	13.63	17.70
(XIV) Elongations, total	1	1.24	1.72	2.37	3.30	4.51

(VII) for any pass=(III) for same pass minus (III) for preceding pass.
(VIII) = (VII) x .99 (IX)=(VIII) / .746 (X) calculated from (I).
(XI) = (X) x (VI) x 12x60x.275/2000 = (X) x (VI)/10 (XII) = (IX)/(XI).
(XIII) for any pass = sum of (XII) for that pass and all preceding passes.
(XIV)=(Initial slab area)/(X).

TABLE II-A

Alternative Method of Calculation of Test Data Given in Table II
Data for Lines I to X are the same as in Table II

	Pass	0	1	2	3	4	5
XV	Reduction in area, sq.in.	....	4.1	4.8	3.44	2.51	1.73
XVI	Length entering pass, feet	....	30.5	37.8	52.3	72.3	100.1
XVII	Cubic inches displaced	....	1500	2175	2155	2180	2090
XVIII	Length after pass, feet	30.5	37.8	52.3	72.3	100.1	137.5
XIX	Seconds per pass	....	24.0	24.0	24.0	24.0	24.0
XX	HP-Seconds per pass	....	9080	14030	14320	15920	15920
XXI	HP-Seconds per cu.in. displaced	....	6.05	6.45	6.65	7.3	7.6

(XVII) = (XV) x (XVI) x 12.
(XX) = (IX) x (XIX).
(XXI) = (XX)/(XVII).

sulting in the graph designated as Curve 1. I find it most convenient to plot these "Power Curves" on semi-logarithmic cross section paper as illustrated, but any ordinary graph paper may be used. If you wish to make use of any of the curves printed in this article I would suggest that you re-plot them on paper having fairly small divisions.

Some users of this same general method prefer to calculate and plot the data in somewhat different form. Table II-A and Curve 1-A illustrate this alternative. Consider pass 2. The reduction in area in this pass is the area after pass 1 minus the area after 2. The length entering 2 is the same as the length leaving 1. Now, we may say that in going through 2 a volume of metal equivalent to the product of the **reduction** in area and the **length entering** the pass is displaced, or shoved off endwise as it were. This amount of metal displaced is shown in line XVII.

The time per pass in seconds is easily calculated, and multiplying this by the horsepower during the pass (line IX of Table II) we have the hp.-seconds per pass. Then dividing the hp.-seconds per pass by the volume of metal displaced gives the hp.-seconds per cu. in. displaced, which may be plotted against elongations as shown in Curve 1-A.

POWER CURVES AND THEIR USE

From the example which has just been given it is hoped the general process of taking power tests and putting the results in usable form is apparent. Obviously there are many minor variations of the method of calculating and plotting the results which may be employed. I have found the curves showing total hp.-hrs. per net ton against total elongations most convenient. Included with this article are several typical curves of this nature.

Now that we have this data, we can use it in determining the power required to drive proposed mills. If we calculate the elongations made in each pass, and the number of tons rolled in each pass, from the proper curve we can determine the horsepower-hours to roll each pass. Then from this and the time for the pass the horsepower during the pass can be calculated.

In order to make the method of determination of motor capacity clear, a concrete problem will be studied.

CALCULATION OF MOTORS FOR HOT STRIP MILL

As an example, let us work through the calculation of the power requirements of a continuous hot strip mill.

Mill

Assume the mill consists of ten stands, individually driven. The first four are two-high with plain bearings, spaced far enough apart so that the same slab is never in more than one stand. The last six are 4-high, with roller bearings, spaced on 18-ft. centers. The range of products, and spacing of stands, is such that constant speed motors may be used for the four 2-high stands, but adjustable speed motors will be needed for the 4-high stands.

Duty

The maximum work on the mill will be rolling 36"x.065" strip, at 1200 FPM, from 36"x3"x7' slabs, at the rate of one slab every 30 seconds.

Calculations

The calculation of the motor outputs is shown in Table III. A large part of this table is self-explanatory.

Columns II and III are, of course, determined from the data supplied by the mill designer.

Column IV is obtained by dividing the original area, 108 sq. in., by the area for each pass. For any pass this gives the total elongation made from the original slab. For example, after pass 5 the area is 10.8 sq. in. The original area, 108, divided by 10.8 gives 10 as the total number of elongations made to and including pass 5. This obviously is also the ratio of the length after the pass to the original length.

Column V contains the values determined from the "Power Curves." As the first four stands of the mill are fitted with plain bearings, and we start with a 3" thick slab, curve 10 fits the conditions very well and can be used for these stands. Obviously, the same curve will not do for the rol-

TABLE III

Calculation of Power Requirements
10-Stand Continuous Strip Mill
Four 2-High, 26" x 44" Stands, Plain Bearings
Six 4-High, 20"-40" x 44" Stands, Roller Bearings
Rolling 36" x .065" strip from 36" x 3" x 7' — 2540 lb. slabs; one slab every 30 seconds.

I Pass	II Area Sq. In.	III Length Ft.	IV Elongations	V HP-Hrs. per Ton Total	VI HP-Hrs. per Ton Per Pass	VII HP-Sec. Per Pass	VIII Roll RPM	IX Ft. Per Sec.	X Sec. Per Pass	XI Net HP Per Pass	XII Friction HP	XIII Motor Output HP	XIV RMS Motor HP
Roughing Stands — 2-High — Plain Bearings — Curve 10													
0	108	7	1										
1	74.0	10.2	1.46	2.2	2.2	10,070	30	3.41	2.99	3370	40	3410	1080
2	48.0	15.7	2.25	5.2	3.0	13,700	30	3.41	4.60	2980	40	3020	1190
3	30.5	24.7	3.54	9.2	4.0	18,300	45	5.11	4.83	3790	60	3850	1550
4	17.8	42.5	6.07	15.7	6.5	29,700	45	5.11	8.31	3570	60	3630	1910
Finishing Stands — 4-High — Roller Bearings — Curve 11													
			6.07	11.5									
5	10.8	70.0	10.0	18.4	6.9	31,600	49.6	4.34	16.2	1950	50	2000	1470
6	6.7	112.8	16.1	28.0	9.6	43,800	80.0	6.98	16.2	2700	60	2760	2030
7	4.5	168	24.0	38.6	10.6	48,500	119	10.4	16.2	2990	70	3060	2250
8	3.4	222	31.7	48.0	9.4	43,000	157	13.7	16.2	2660	80	2740	2020
9	2.76	274	39.1	55.8	7.8	35,700	194	16.9	16.2	2200	90	2290	1680
10	2.34	324	46.2	62.5	6.7	30,700	229	20.0	16.2	1890	100	1990	1460

For any pass:
(IV) = Original area divided by area after pass.
(V) = Obtained from Power Curves No. 10 and 11.
(VI) = (V) minus (V) for preceding pass.
(VII) = (VI) x 1.8 x 2540 (weight of slab).
(VIII) = Mill data.

$$(IX) = \frac{(VIII) \times 3.1416 \times \text{Roll diam.}}{60 \times 12} = \frac{(VIII) \times \text{Roll diam.}}{229}$$

(X) = (III)/(IX).
(XI) = (VII)/(X).
(XII) = Estimated.
(XIII) = (XI) + (XII).

$$(XIV) = \sqrt{\frac{(XIII)^2 \times (X) + (XII)^2 \times (30-(X))}{30}}$$

ler-bearing finishing stands, and for these we will select Curve 11. We start on this curve at the elongations after pass 4, namely 6.07 and carry through to the finish.

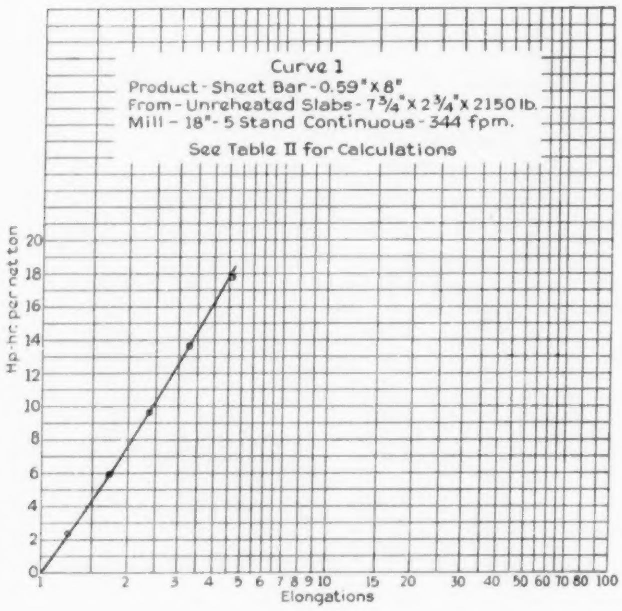
Now for any pass column V gives the total power consumption up to and including that pass. (This is not strictly true for the six finishing passes in this case, as the values here show what the total power would be if the roughing stands also had roller bearings.) But, what we are most interested in is the work for each pass. This is given in column VI, in terms of horsepower-hours per ton of steel rolled. For any pass it is obtained by subtracting from the total (column V), the total for the preceding pass. For example, for pass 8, 48.0 (total work through pass 8) minus 38.6 (the total work through pass 7) gives 9.4 horsepower-hours as the amount of work necessary to roll one ton of steel through the pass.

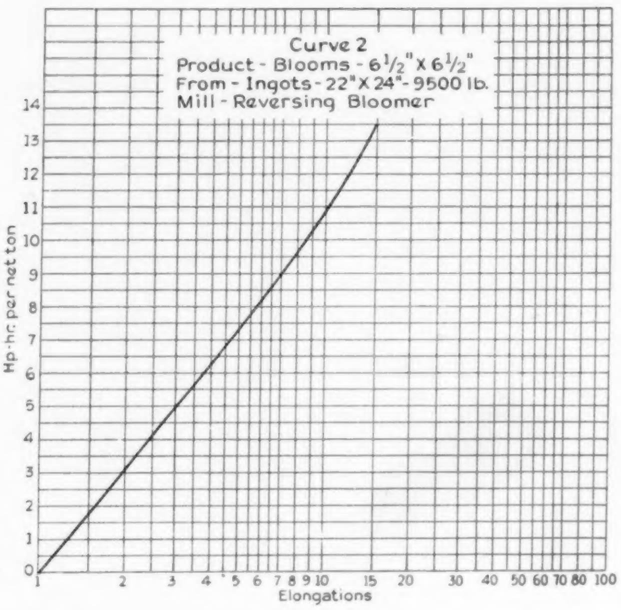
To get the work per pass into more usable terms, we convert it into horsepower-seconds per pass in column VII. Note carefully this difference between VI and VII. For any pass VI gives work **per ton,** but VII gives work **per pass.** In other words, VII takes into account the **tons per pass,** and gives the work per pass for **each piece.** To show how this is obtained, consider pass 8. The work done in this pass is 9.4 hp.-hrs. per ton. To convert this to hp.-seconds per ton it is only necessary to multiply by 3600, the number of seconds in an hour. Therefore, we have 9.4 x 3600 = 33,840 hp.-sec. per ton. Now the slab weighs 2540 lb., or 2540/2000 = 1.27 net tons. Therefore, the horsepower-seconds per pass per slab is 33,840 x 1.27 = 43,000, as given in column VII. It will be noted, for the general case, that what we have done is this:

$$(VII) = (VI) \times 3600 \times \frac{\text{weight}}{2000}, \text{ or } (VI) \times \text{weight} \times 1.8$$

The roll speed, VIII is specified by the mill designer, and from it and the roll diameter it is a simple matter to calculate the delivery speed for each pass, as given in column IX.

Knowing the delivery speed of each pass, and the length after each pass, we calculate the number of seconds required for each pass by dividing the length, III, by the speed in feet per second, IX, and obtain the values in column X.





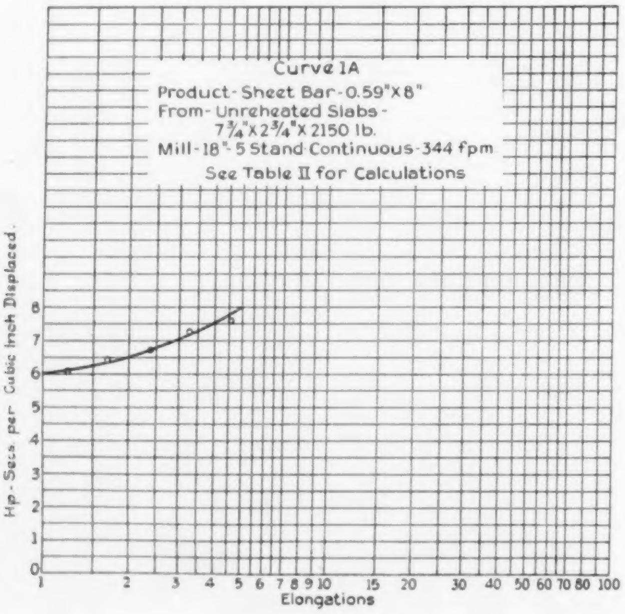
Now we have the horsepower-seconds work for each pass in column VII, and the seconds for each pass in X. Dividing horsepower-seconds by seconds ought to leave just horsepower, and this is exactly what we do, with the result that in XI we have the net horsepower required during the rolling of each pass. You will remember that the values on our power curves do not include the no-load friction losses in mill, pinions, etc., so we must estimate these, in column XII, and add them to XI to obtain XIII, the calculated output from the motor shaft during the rolling of each pass.

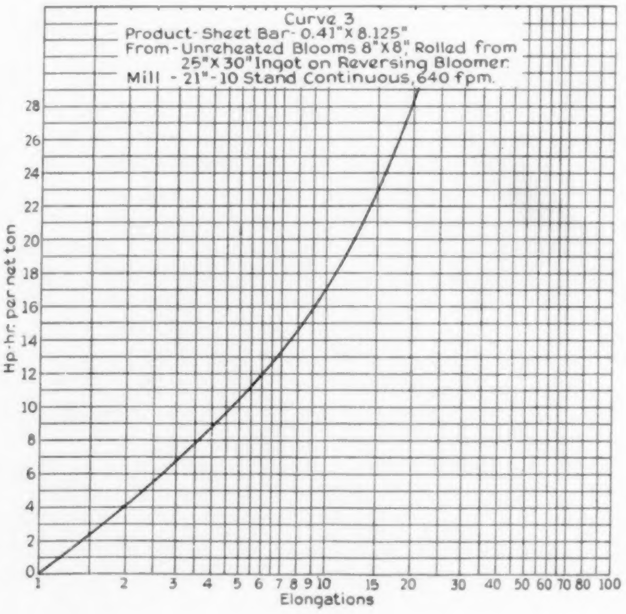
Selection of Motors

The output of each motor during the rolling of each pass is given in column XIII. If the stands were to run 100% full we would simply need to select motors whose normal ratings are enough above the figures in this column to give a reasonable factor of safety. However, as the maximum output of the mill is one slab every 30 seconds, and the maximum length of pass is 16.2 seconds, it is probable that we can use motors whose ratings are less than the figures in XIII. To determine this it is necessary that we know the heating effect of the load on each motor, or in other words the root-mean-square (rms.) of the load.

Consider pass 8. The load while metal is in the rolls is 2740 hp. for 16.2 seconds. For the remainder of the 30 seconds allotted to each slab, or 13.8 seconds, the load is the friction, 80 hp. To calculate the rms.-hp. we proceed as follows:

$$\begin{array}{rr} (2740)^2 \text{ x } 16.2 = & 121,623,120 \\ (80)^2 \text{ x } 13.8 = & 88,320 \\ \hline 30.0 & 121,711,440 \text{ hp.}^2\text{sec.} \end{array}$$





$$\frac{121{,}711{,}440}{30} = 4{,}057{,}048 \text{ mean hp.}^2$$

$$\sqrt{4{,}057{,}048} = 2020 \text{ rms.-hp.}$$

In other words, the heating of a motor on this intermittent duty cycle will be about the same as though a constant load of 2020 hp. were applied.

The r.m.s. of the duty cycle on each motor is given in column XIV.

Now we are ready to decide on the required motor ratings, but we must proceed very cautiously, and take account of many things. It is not enough to simply select a motor whose rating is slightly higher than the r.m.s. of the duty cycle. Such a motor might not have sufficient overload capacity to carry the actual load during the pass. We must also consider the matter of possible duplication of motors on the different stands, in order to reduce the amount of spare parts required.

Let us consider first the motors for the roughing stands. As these are to run at constant speeds, either synchronous or induction motors can be used. As the starting torque required for a single stand is low, synchronous motors should be employed unless we wish to add flywheels to the drive in order to reduce the peak loads on the power system, or to permit somewhat smaller motors to be used.

For pass 1 we have a maximum load of 3410 hp. and an r.m.s. of 1080 hp. Now while what may be termed "standard mill type" synchronous and induction motors are usually designed to give a maximum pullout torque at least 250% of normal, I do not consider it good practice to let the calculated peak load be more than approximately 175% of normal. This leaves a factor of safety to take care of cold steel, incorrect drafting and other things that are sure to occur. So, if we use no flywheel, the normal motor rating should not be less than about 3410/1.75 = 1950 hp. To use the nearest standard rating we should say that a 2000 hp. synchronous motor would do the job.

If an induction motor with flywheel were employed, a smaller motor could be used, but an induction motor is inherently more costly than a synchronous, the wheels cost money, and so does the slip control apparatus, whether it be magnetic or liquid slip regulator. Experience has shown that it does not pay to use a wheel on a main drive when the peak is more than 3 or 4 seconds long. On this particular stand there is not much difference in cost between the **complete** induction motor drive and the synchronous. However, we must look ahead a little. In stand 2 the pass is so long that it does not seem wise to consider wheels. Here we need a motor to carry 3020 hp. peaks. 3020/1.75 = 1725 hp. Therefore, we might use a 1750 hp. motor, but if we make motor 1 synchronous we can to advantage use duplicate 2000 hp. synchronous motors on stands 1 and 2, thus saving in spares required. If these motors are 300 r.p.m. machines a single-reduction gear can be used with each, having a ratio of 10:1, which seems to be a fair maximum value for mill type single-reduction gear units.

For stand 3, to carry the peak, we need a normal rating of at least 3850/1.75 = 2200 hp.

For stand 4 we need at least 3630/1.75 = 2080 hp. Probably the best thing to do is to use two duplicate 2250 hp. synchronous motors for 3 and 4. If we have a sixty cycle supply we can make them 450 r.p.m. and use 10:1 gear sets duplicates of those on stands 1 and 2.

You will note that the ratings of all of the motors so far selected are well above the r.m.s. load values. That is, for these roughing stands the motor rating is determined by the peak load rather than by the r.m.s.

Now let us turn to the d-c. motors for stands 5 to 10. To start with we will lay down certain principles. So-called "standard mill type" adjustable speed d-c. motors having a speed range of 2:1 by field control are usually designed to carry momentary maximum loads equivalent to 200% of normal rating. Of course, the maximum load a d-c. motor can successfully carry is limited by commutation. A motor may carry 200% load for short periods occasionally with no difficulty, but if this load is very frequently repeated trouble with commutator and brushes may result, particularly if the motor is operating at or near its maximum speed. Therefore, I consider it good practice to limit the calculated peak loads to not over 150% of normal at double speed, and 175% at basic speed. If the motor operates over a range of more than 2:1 by field control the allowable loads at more than double basic speed should be still further decreased. It is apparent that for all of stands 5 to 10 the maximum load is only about 135% of the r.m.s., hence the r.m.s. becomes the determining factor.

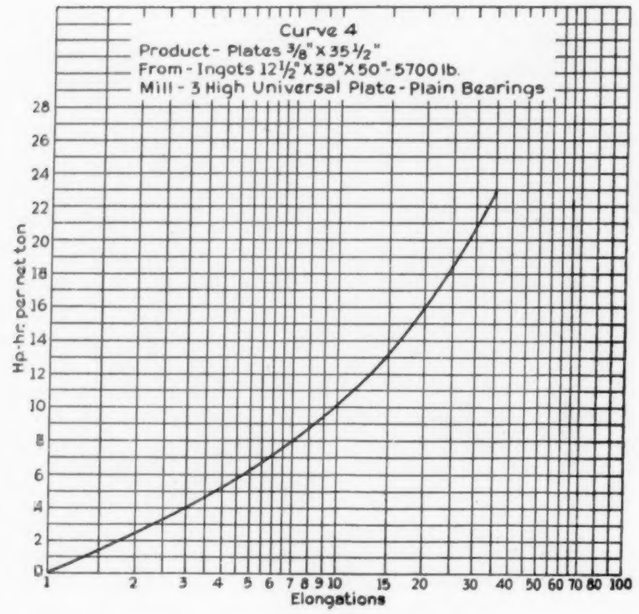
From inspecting the values in column XIV it looks as though we had better use 2250 hp. motors for stands 6, 7, and 8, making them all duplicates. For stand 9 we need at least 1750 hp., while 1500 may do for 5 and 10. However, to permit using duplicate motors, and provide a little excess capacity which may prove useful, it is probable that 1750 hp. would be selected for each of 5, 9, and 10. Then these three could be made duplicates.

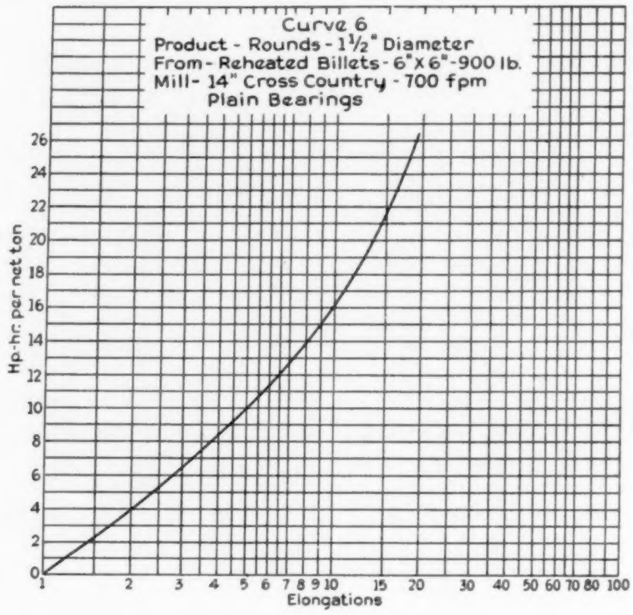
Of course, all of the foregoing selections are on the assumption that no other proposed rolling schedule shows a greater load on any of the motors than the one calculated. All schedules must be carefully checked, and motors chosen capable of carrying the heaviest loads encountered.

As to desirable speeds for the d-c. motors, we must watch that we do not get too high a gear ratio, nor an undesirably high motor speed.

Table V gives what may be considered as conservative values of maximum speeds for the various standard horsepower ratings for 600 volt d-c. mill motors with speed ranges by field control of 2:1 and 3:1. The values in this table can be, in most cases, and have been in quite a few cases, exceeded somewhat when conditions make it desirable to obtain higher speeds, but they represent good, safe upper limits. In the same way a gear ratio of 10:1, for large units, is usually considered about the maximum desirable for a single reduction gear, but this also can be and has been exceeded when conditions make it necessary.

Assume that the mill designer says he needs the following range in roll speeds to take care of all products:



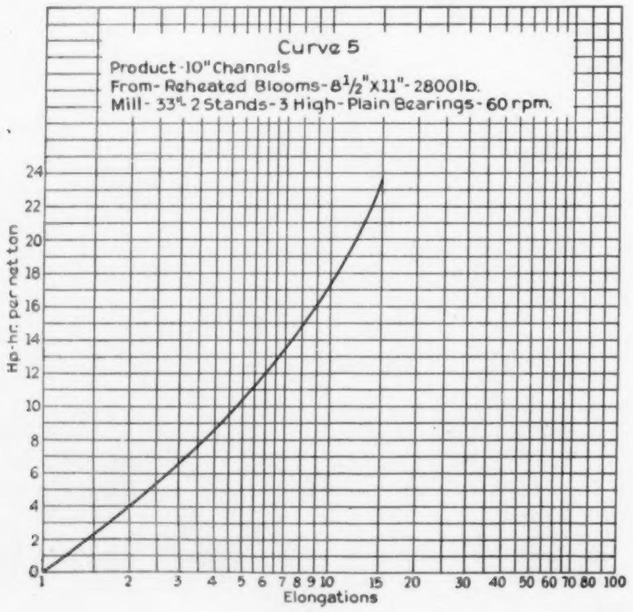


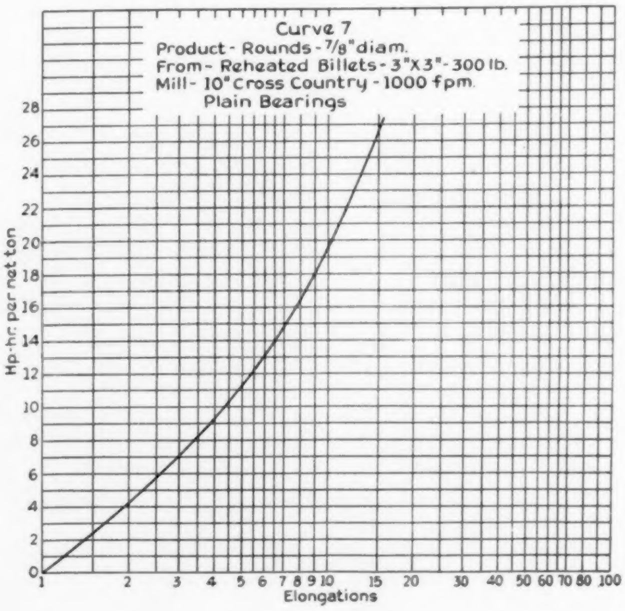
Stand 5 — 30 to 60 r.p.m.
Stand 6 — 48 to 96 r.p.m.
Stand 7 — 68 to 136 r.p.m.
Stand 8 — 85 to 170 r.p.m.
Stand 9 — 107 to 214 r.p.m.
Stand 10 — 130 to 260 r.p.m.

We can make the 2250 hp. motors on stands 6, 7, and 8, 225/450 r.p.m., which is a safe maximum speed for a motor of this capacity. This will give gear ratios of 4.69, 3.31, and 2.65, which are satisfactory. The 1750 hp. motors for 5, 9, and 10 can be 250/500 r.p.m., giving gear ratios of 8.33, 2.33, and 1.92. One may ask why the last two motors should not be direct connected, and this is a very sensible question. A check will show, however, that a saving of a few thousand dollars can be made on each of these drives by using the gear and higher speed motor. Usually, if a gear ratio of about 2 to 1 can be obtained, the geared drive will be lower in first cost. Also it permits of using duplicate motors. On larger mills, with motors of greater capacity, it would often be difficult to get direct connected motors on sufficiently close centers, because of the large size of slow speed machines.

So now our drives line up as follows:

Stand	Gear Ratio	Motor HP	Motor RPM	
1	10.00	2000	300	Synchronous
2	10.00	2000	300	Synchronous
3	10.00	2250	450	Synchronous
4	10.00	2250	450	Synchronous
5	8.33	1750	250/500	Direct current
6	4.69	2250	225/450	Direct current
7	3.31	2250	225/450	Direct current
8	2.65	2250	225/450	Direct current
9	2.33	1750	250/500	Direct current
10	1.92	1750	250/500	Direct current





It is evident that duplication has been carried as far as feasible, both as to motors and gear units.

Selection of Motor-Generator

Having determined the motors to use, it becomes necessary to decide on the capacity of the motor-generators to furnish power to the d-c. drives. The r.m.s. load on the generators will obviously be less than the sum of the r.m.s. loads on the motors. Neither will the peak be equal to the sum of the peaks unless the metal will be in all stands at the same time. We must find out.

We have the delivery speed of each stand, and we know the spacing is 18 ft., so we can calculate the time required for the front end of the slab to pass from each stand to the next, as follows:

Stand	Time for Metal to Travel to Next Stand
5	4.14 sec.
6	2.61 sec.
7	1.73 sec.
8	1.31 sec.
9	1.07 sec.

Knowing that it takes the slab 16.2 seconds to go through each stand, we can readily plot the load cycle on the generators from this data and the motor loads in columns XI and XII. We must remember that we have the friction loads on all motors all the time. From this duty cycle, Fig. 2, we calculate the r.m.s. of the combined motor loads to be 9590 hp., and the peak 14,840 hp. Converting into kilowatts, and allowing about 8% for motor losses, the r.m.s. load on the generators will be 7780 kw., and the peak load 12,000 kw.

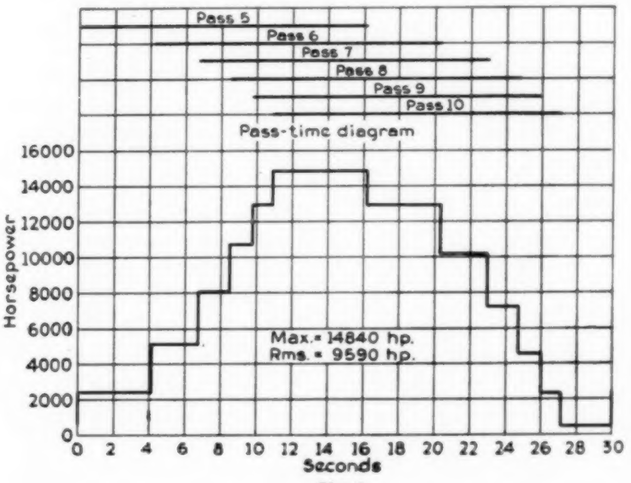


Fig. 2
Combined Load on D-c. Motor on Stands 5-10 Inclusive

In choosing generators I believe it is best to limit the calculated maximum load to about 150% of the normal rating. This then allows some excess capacity to take care of cold steel, etc. In this particular case it is evident that 8000 kw. generator capacity will just do the trick, and if we are sure that the schedule calculated is the absolute maximum the mill will be called on to do, it is safe to plan on using two 4000 kw. sets. If we are not so sure, then we might better put in two 5000 kw. sets, or possibly one 9000 kw. The 4000 and 5000 kw. sets would each consist of two generators and a synchronous motor. The 9000 kw. unit would have three generators and a motor.

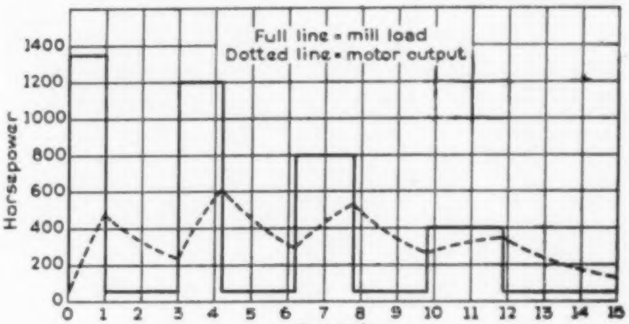


Fig. 3
Duty Cycle of Roughing Stand, Showing Effect of Flywheel in Reducing Peak Load on Drive

The 9000 kw. set might be somewhat less costly than two 5000 kw. units, but obviously does not have the same flexibility in operation. If two 5000 kw. sets were installed, probably a great number of schedules could be rolled with only one set in use. Also, two units are more reliable than one. I would be inclined to favor either the two 4000 kw. or two 5000 kw. sets.

I hope that I have succeeded in making clear the general method employed in laying out the drive for a continuous mill.

CALCULATION OF MOTORS FOR OTHER TYPES OF MILLS

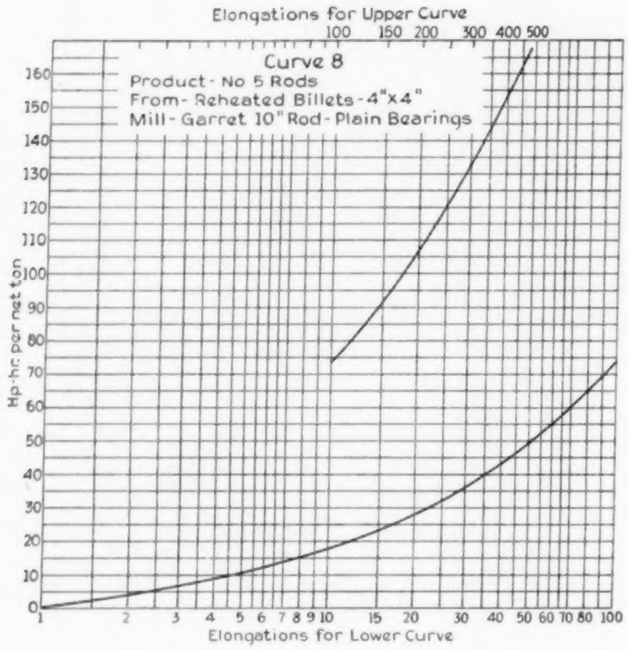
It is obvious that the general principles of calculating the load imposed on the drive, exemplified in the preceding paragraphs, will apply to practically any type of mill. Modifications in details must be made to fit various conditions.

For example, where several stands are driven by one motor, the load due to each pass is determined in the same general way, but care must be used in combining these loads, taking account of intervals, pass times, and so on, to arrive at the total duty on the drive. It is often convenient to plot the duty cycle, just as the duty on the generators for the hot strip mill was plotted in Fig. 2. Then calculation of the maximum peak horsepower and the r.m.s. of the cycle will permit the motor to be selected.

Drives With Flywheels

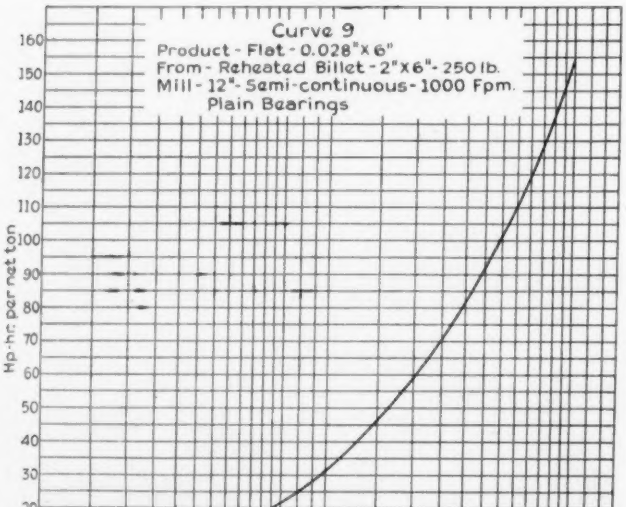
As previously noted, it is not usually economical to attempt to use flywheels to reduce the peak load on the motor if the peak lasts more than about three seconds. Obviously, neither is it desirable to employ a wheel if the ratio of the peak load to the r.m.s. load is not more than 1.5 or 2. But if the load cycle does have high peaks, of short duration, the use of a wheel may prove advantageous.

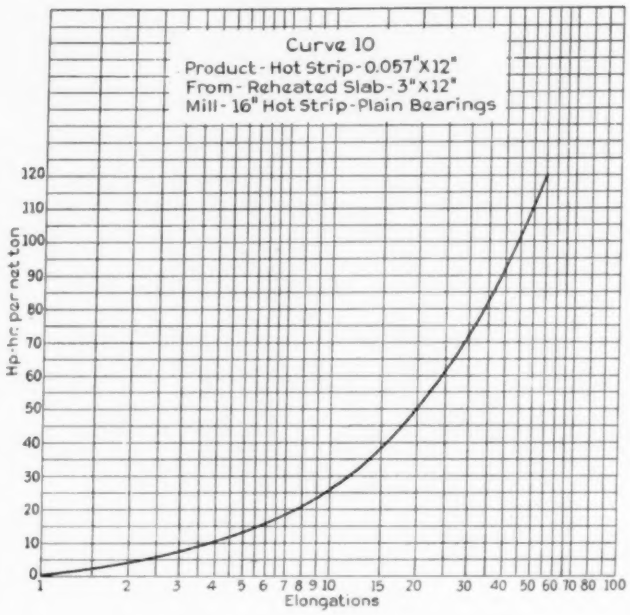
Flywheels can be used to best advantage with induction motor drives, on mills operating at constant (approximately) speed. A flywheel on an adjustable speed drive is not a very satisfactory combination. The energy stored in the wheel varies as the square of the r.p.m. Therefore, on a drive with 2:1 range the wheel will have four times the stored energy at top speed that it does at basic.



It seems invariably to be the case that where we have an adjustable speed drive on which the possibility of using a flywheel looks attractive, we find that the highest peak loads occur at the low speed, where a hp.-sec. of flywheel energy costs about four times as much as it would if the wheel could run at its maximum safe speed. Furthermore, the heavy wheel makes it difficult to adjust the motor speed quickly. Also, to permit any utilization whatsoever of its energy, the wheel must slow down during the load period, which precludes its use on strictly continuous mills. So we usually end up by leaving the wheel off from an adjustable speed drive.

In making calculations on a mill which will use wheels, the load cycle of the mill is determined just as has been described and illustrated, but from that point on the process of selecting the motor and wheel is somewhat more complicated.

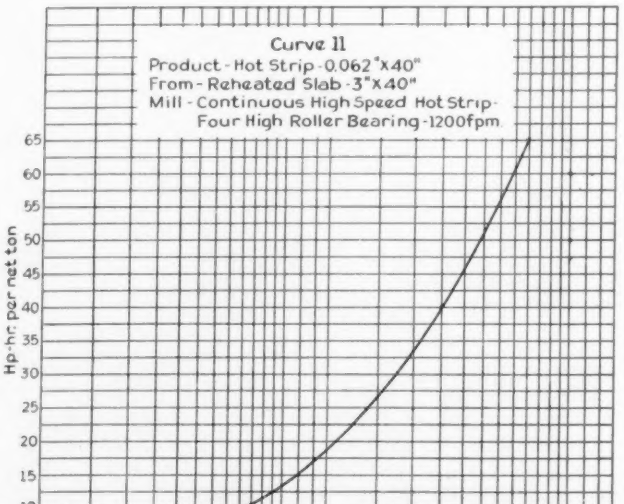




Assume that we have a roughing stand, constant speed, on which the load due to the mill is as shown in Fig. 3. If no wheel were used the **r.m.s.** of the cycle would be as follows:

$$\begin{aligned}
\text{Pass 1} &— (1350)^2 \times 1 &= 1{,}822{,}500\\
\text{Pass 2} &— (1200)^2 \times 1.2 &= 1{,}728{,}000\\
\text{Pass 3} &— (800)^2 \times 1.6 &= 1{,}024{,}000\\
\text{Pass 4} &— (400)^2 \times 2.1 &= 336{,}000\\
\text{Intervals} &— (50)^2 \times 9.1 &= 22{,}750\\
& \quad\quad\quad\quad\;\; 15 & 4{,}933{,}250
\end{aligned}$$

$$\sqrt{\frac{4{,}933{,}250}{15}} = 572 \text{ rms.-hp.}$$



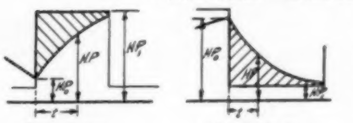
The **average** of the cycle is:

Pass 1 — 1350 x 1 = 1350
Pass 2 — 1200 x 1.2 = 1440
Pass 3 — 800 x 1.6 = 1280
Pass 4 — 400 x 2.1 = 840
Intervals — 50 x 9.1 = 455
5365

$$\frac{5365}{15} = 357 \text{ average hp.}$$

TABLE IV.

FLYWHEEL CALCULATIONS



HP_a = RATED HP OF MOTOR
S_a = SLIP AT HP_a (Expressed as a Decimal)
E = TOTAL HP-SEC. OF STORED ENERGY IN FLYWHEEL AT SYNC SPEED $= \frac{Wr^2 (R.P.Sec.)^2}{838}$

$$A = \frac{0.5\, HP_a}{E\, S_a}$$

Pass No	Elapsed Time Total	Elapsed Time Period	At	ε^{At}	HP_o	HP_1	HP_1-HP_o	$\frac{HP_1-HP_o}{\varepsilon^{At}}$	$HP = HP_1 - \frac{HP_1-HP_o}{\varepsilon^{At}}$
1	.5	.5	.208	1.22	50	1350	1300	1065	285
	1.0	.5	.208	1.22	285	1350	1065	873	477
	2.0	1.0	.416	1.51	477	50	-427	-283	333
	3.0	1.0	.416	1.51	333	50	-283	-187	237
2	3.5	.5	.208	1.22	237	1200	963	790	410
	4.2	.7	.281	1.32	410	1200	790	598	602
	5.2	1.0	.416	1.51	602	50	-552	-365	415
	6.2	1.0	.416	1.51	415	50	-365	-241	291
3	7.0	.8	.333	1.39	291	800	509	366	434
	7.8	.8	.333	1.39	434	800	366	263	537
	8.8	1.0	.416	1.51	537	50	-487	-323	373
	9.8	1.0	.416	1.51	373	50	-323	-214	264
4	10.9	1.1	.459	1.57	264	400	136	87	313
	11.9	1.0	.416	1.51	313	400	87	57	343
	13.0	1.1	.459	1.57	343	50	-293	-187	237
	14.0	1.0	.416	1.51	237	50	-187	-124	174
	15.0	1.0	.416	1.51	174	50	-124	-82	132

Now by using flywheels, the peak loads on the motor will be below the mill peaks, so the r.m.s. on the motor will be below the value of 572 hp., but not so low as the average of 357 hp. unless we could perfectly equalize the cycle, making the load on the motor absolutely uniform. Even then we would need a motor somewhat larger than 357 hp. to allow for the slip resistor losses which must pass through the motor. So in order to have some margin of safety above the average value, suppose we see how a 500 hp. motor will do.

As a basis for assuming a wheel, experience shows that a suitable wheel for a merchant mill will usually have a stored energy at synchronous speed, expressed in horsepower seconds, of ten to fifteen times the figure representing the normal horsepower rating of the motor. For sheet mills this ratio may be on the order of 20, and for plate mills 15. So suppose we try a wheel with 12 x 500 = 6000 hp.-sec. stored energy.

In calculating the output of a motor equipped with a flywheel use is made of a formula developed by Mr. F. G. Gasche. A modification of this formula is shown at the top of the chart, Table IV. For the particular case under discussion HPa = 500. Sa we will assume to be .10, or in other words there will be enough resistance in the rotor circuit to give 10% slip. E = 6000

$$\text{Therefore } A = \frac{0.5 \times 500}{6000 \times 0.1} = 0.416$$

The calculations in Table IV are fairly self explanatory. First we select points, along the time scale of the duty cycle, at which we wish to calculate the motor load. These are given in the column headed "Elapsed Time, Total." Points corresponding to the beginning and ending of each pass should be chosen, with enough in between to give the shape of the curve. Then the increments of time are given in column "period t."

Column At is merely the product of the constant A, which has been calculated to be 0.416, and the "period t." E^{AT} is obtained from the chart, Fig. 4. E is the base of the Naperian system of logarithms, 2.71828. HPo is the output of the motor at the beginning of "Period t." HP_1 is the load due to the mill, or the driven load. HP is the output of the motor at the end of "Period t." It is obvious that this becomes the HPo of the next "Period t." Care must be used to keep the algebraic signs correct in making the calculations in the last three columns.

The values in the last column of Table IV are plotted against time, in dotted lines, on Fig. 3; these dotted lines show the approximate horsepower output of the motor. The results are not strictly accurate, for evidently the time of each pass will be slightly longer than shown, because the motor must slow down to let the flywheel give up some of its energy. Since the resistance in the motor rotor circuit is sufficient to give 10% slip at full load, the slip of the motor at any point, in percent, will equal 0.1 times the motor output, in percent of normal rating, at that point.

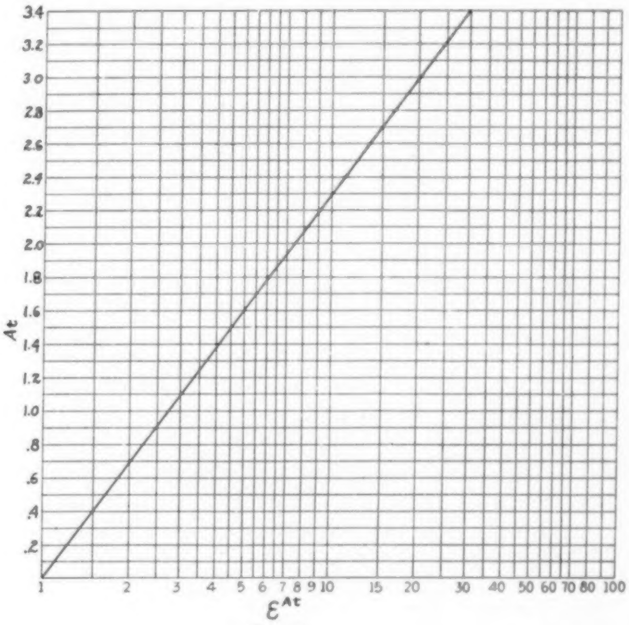


Fig. 4

To make the duty cycle more strictly accurate we may now calculate the average speed during, and therefrom the time for, each pass based on this approximate duty cycle. We should also make the initial load, at the beginning of pass 1, correspond to the calculated output at the end of the cycle.

However, such refinements are rarely necessary for practical purposes, and we will proceed on the basis of the approximate duty cycle.

First the r.m.s. of the output curve may be calculated by the step-by-step method. Where a sloping **straight** line represents the output during the time t, if A is the output at the beginning of t, and B at the end, then the hp.^{2}t for this interval is equal to

$$\frac{(A^2 + B^2 + AB)}{3} \times t$$

For example, the output during the first pass is practically a straight line, so for this pass

$$\text{hp.}^2\text{t} = \frac{(50^2 + 477^2 + 50 \times 477)}{3} \times 1 = 84{,}430 \text{ hp.}^2\text{sec.}$$

So we proceed, taking increments of time, and then summing up all the values of hp.^{2}sec., dividing by the total time, 15 seconds and extracting the square root, we find the rms. of the output is approximately 372 hp. Now the "heating" load on the motor will be this plus about 10% for losses in the slip resistor, or a total of 409 hp. The maximum peak is 602 hp. The effect of the wheel in reducing not only the peak load, but the rms. on the motor as well, is at once evident.

It is apparent that the combination of motor and wheel we have started with is somewhat larger than necessary. However, the next smaller standard motor rating is 400 hp., which is too small. If the load cycle calculated can be accepted as the absolute maximum, we might try reducing the wheel to 5000 hp.-sec. and go through the calculations again.

If instead of the fixed slip resistor a liquid slip regulator is used, absolutely limiting the output of the motor to a predetermined value, the calculations will be somewhat different. If it is assumed that the slip regulator can reduce the rotor resistance to zero, which of course is not correct, then the motor output curve becomes a series of vertical and horizontal lines. For example, assume the regulator is set at 750 hp. When pass 1 begins the load would immediately rise to 750 hp., and stay there until the motor has delivered a number of hp.-sec. equivalent to the amount required by the pass and the friction load, and then immediately drop to 50 hp. Actually there will always be from 2% to 4% resistance in the rotor circuit, so the output curve will rise and fall in steep curves, calculated as in Table IV, and be horizontal between.

Reversing Mill Drives

The method of calculating the motor capacity required on a reversing mill drive is too long to be covered within the space of this article. It has already been so thoroughly and clearly described by Mr. L. A. Umansky in his paper "Selection of Electric Drives for Reversing Mills" pages 327 to 349 of the Proceedings of the Association of Iron and Steel Electrical Engineers" for 1925, that I will merely suggest that if you are interested you should read Mr. Umansky's treatise on this subject.

GENERAL CONSIDERATIONS

I think it will be evident that in determining the motor capacity required for a mill drive, the most important consideration is the selection of the proper "power curve."

In the first place the curve selected should be one which starts with a bloom or billet of the same, or very nearly the same, size as is to be used in the mill under consideration. It would not do to use Curve 6, starting with a 6"x6" billet, to calculate the power needed to roll rods starting from a 4"x4" billet. At 20 elongations on the curve, the area of the 6"x6" billet has been reduced to only 1.8 sq.in. At this same 20 elongations the area of the 4"x4" billet has been reduced to 0.8 sq.in. Even though the two billets started at the same temperature, it is obvious that the 0.8 sq.in. rod would be considerably colder than the 1.8 sq.in. rod. Temperature has a tremendous effect on the amount of power required to roll. Therefore, the use of this curve would result in too small a motor. This is readily apparent from a comparison of the ordinates of curves 6 and 8.

It is possible to use "power curves" which do not quite fit the case by making some corrections. Suppose we want to start with a 5½"x5½" billet. We might make use of curve 6 in this way. The initial area on curve 6 is 36 sq.in. The initial area of the 5½" billet is 30.25 sq.in. This area, therefore, occurs at 36/30.25, or 1.19 elongations on Curve 6. Now the change in temperature in a fairly large billet is not great in the first pass or two, so without too much error we may in this case assume that the temperature corresponding to 1.19 elongations on Curve 6 is the same as the initial temperature of the 5½" billet, assuming that both are originally heated to ordinary rolling temperatures.

Therefore, for one elongation on the 5½" billet, we will start at 1.19 on curve 6, and read the corresponding hp.-hrs. For two elongations of the 5½" billet we will read the curve at 1.19x2, or 2.38 elongations, and so on. It is apparent that in this way we are working with equal **areas,** and assuming that this means equal temperatures. It is obvious that this will not hold for any great difference in original billet sizes. It is also obvious that whatever error there is will result in too high a value of power; in other words, it is on the safe side.

In the second place, the finished shape and size to which the curve applies must be similar to that under consideration. It takes more power to roll a structural shape, or a rail, than to roll a round or square bar. There is more deformation of the metal for a given number of elongations, and there is more friction between the flanges of the piece and the grooves in the rolls.

It takes more power to roll a flat, from a given billet, than to roll a round, because the flat has more area in contact with the rolls and cools faster.

In the third place, the "power curve" should have been taken from data on a mill of the same general type as the one under consideration. A rod will finish hotter from a high speed continuous mill than from a hand looping mill.

Roller bearings on the roll necks make a large difference in power requirement, as is apparent from a comparison of Curves 10 and 11. The reduction in power due to the use of roller bearings may range from 10% to 60%, depending on the roll neck pressure.

In the fourth place, and this is very important, the initial temperatures must correspond. If we want to take a 6"x6" billet direct from a bloomer and roll it without reheating, it would not do to use Curve 6, as this is based on starting with a reheated billet. A difference of only about 200 degrees F. in temperature may double the power required.

The matter of temperature is unquestionably the most important factor in determining the work required to roll a ton of steel to a given number of elongations, and yet it is one on which it is difficult to get accurate information. It is also one of the most variable factors. At the start of a turn, with the furnaces full, the steel may be very considerably hotter than after an hours rolling. I have seen many graphic wattmeter records which testify to this fact, the power for a given schedule gradually creeping up and up as colder and colder steel comes to the mill.

The few foregoing paragraphs emphasize the fact that great care must be used in selecting the "Power Curve" to form the basis of a mill drive calculation. They also emphasize the desirability of having available just as much data of this character as possible. It is hoped that steel mill operating engineers may find time to make simple power tests on their various mills, and to publish the data for the good of the industry at large.

CONCLUSION

The general method which has been described, for determining motor capacities for main roll drives, is based fundamentally on a rational comparison of the work to be done on the mill under consideration with the work done, and the power required, on existing mills of the same general character. It perhaps is not so highly theoretical and mathematical as some other methods, but it has proven thoroughly practical.

There are at present installed on main roll drives over 2,000 motors, of 300 hp. and above, totalling more than 2,600,000 hp. I think I am safe in saying that the general method herein described, or some slight modification of it, has been used in calculating the capacity required on the big majority of these drives. Considering the impressive total number, I think that the number of errors which has been made does not bulk very large.

If this article assists the steel mill engineers of this country to make estimates of the power required to drive proposed mills, or if it encourages them to collect and publish power data, or if it gives them a better understanding of the information which an electrical manufacturer needs in order to make the detailed calculations for a main roll drive, it will have accomplished its purpose.

APPENDIX

Miscellaneous Data and Formulae

Table V gives a list of suggested maximum speeds for adjustable speed d-c motors and also for a-c. units. In most capacities higher speed machines can be built, but very often the first cost of such motors would be greater than for the listed speeds.

TABLE V

Suggested Conservative Maximum Speeds for Main Roll Motors

(Motors of higher speeds in most capacities can be, and in some have been, built, where some peculiar local condition requires. The values given should be regarded as guides, rather than absolute limits.)

Horsepower Rating 40 deg. C	600 V. Direct Current Motors Speed Range by Field Control		Synchronous or Induction Motors
	2:1	3:1	
300	700/1400 rpm.	400/1200 rpm.	1200 rpm.
350	600/1200	350/1050	1200
400	600/1200	300/900	1200
500	600/1200	300/900	1200
600	500/1000	300/900	1200
700	500/1000	300/900	1200
800	450/900	250/750	1200
900	400/800	250/750	1200
1000	400/800	250/750	900
1250	350/700	200/600	900
1500	300/600	175/525	900
1750	250/500	150/450	900
2000	250/500	125/375	750
2250	225/450	125/375	750
2500	225/450	100/300	750
3000	200/400	100/300	750
3500	175/350	80/240	600
4000	150/300	70/210	600
4500	125/250	60/180	600
5000	125/250	50/150	600

Symbols

L—length of metal, in feet.
Lo—length before rolling.
Ln—length after pass n.
A—cross-sectional area of metal in sq.in.
Ao—Area before rolling.
An—Area after pass n.
Sn—Lineal speed of metal in fpm. from pass n.
a—acceleration or deceleration in rpm. per sec.
WR^2—moment of inertia in lb.ft.^{2}.
T—torque in lb.ft.
H—Net rolling work in hp.-hrs. per ton.
D—Effective roll diameter in inches.
En—Elongations after pass n.

$$LoxAo = LnxAn \quad \text{.......... (1)}$$

$$T = \frac{WR^2 \times a}{307.6} \quad \text{.......... (3)}$$

$$T = H \times A \times D \times 136 \quad \text{.......... (4)}$$

$$En = \frac{Ao}{An} = \frac{Ln}{Lo} \quad \text{.......... (5)}$$

$$\text{H.P.} = \frac{\text{T x RPM}}{5250} \quad \text{......(6)}$$

$$\text{HP-Sec. in flywheel} = \frac{\text{WR}^2 \text{ x (RPM)}^2}{3{,}225{,}000} \quad \text{......(7)}$$

$$\text{Net tons per hour, pass 100\% full} = \frac{\text{An x Sn}}{10} \quad \text{...(8)}$$

Derivation of (4)

$$\text{H.P.} = \text{H x (Tons per hour)}$$

$$= \text{H x} \frac{\text{A x D x 3.1416 x RPM x 60 x .275}}{2000}$$

$$= \frac{\text{H x A x D x RPM}}{38.6}$$

From (6)

$$\text{H.P.} = \frac{\text{T x RPM}}{5250}$$

$$\text{Therefore} \frac{\text{T x RPM}}{5250} = \frac{\text{H x A x D x RPM}}{38.6}$$

Hence T = H x A x D x 136.

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"Roll Neck Lubricants and Lubrication"

By W. D. HODSON
President, The Hodson Corporation,
Chicago, Ill.

To be presented before A. I. & S. E. E. Annual Convention in Pittsburgh, June 20-23, 1932.

W. D. HODSON

While our Association is chiefly for the benefit of, and supported largely by Steel Plants, directly and indirectly, those most interested personally in our subject are the mill operators. I speak primarily from their viewpoint, with the hope, however, that others concerned; machinery builders, the manufacturers of lubricants and those who have to do with their sale and service, will profit.

This presentation consists of data, statistical and operating methods gathered from engineers and operating officials, many of them members of this Association. Having taken an active part with machinery designers and builders, mill men and operating officials during the past ten years in the improvement of lubrication—lubricants, methods and means of applying—and having developed new types of lubricants designed especially for roll neck lubrication, we will refer to them briefly.

Our main object shall be to present matter that will be informative and constructive, endeavoring to avoid offering anything that might cause controversies unless through them there is promise of mutual benefit to those primarily concerned.

1. Those out in the mills who have to keep them running.
2. The mill owners.
3. The machinery designers and builders.

4. Those dealing with lubricants, without which we would not get very far.

Bearings are important factors of lubrication. We shall discuss them in the order of their importance as determined by the extent of their use.

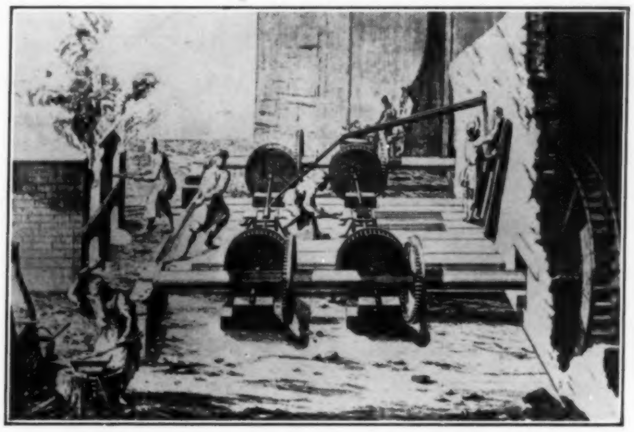


FIG. 1.

1. Standard babbit and bronze bearings.
   (a) Open neck type.
   (b) Full sleeve type.
2. Roller bearings.
3. Composition bearings.
4. Hot neck bearings. (While of standard open neck type, they are a different class pertaining to lubrication).

Each type will be covered where data is available so far as is practical as to: (a) design; (b) lubricants; (c) method of applying; (d) power costs; (e) cost of repairs and maintenance, and (f) life-tonnage.

The method of applying lubricants, stated chronologically are: paddle packed, bags, boxes, oil cups, circulating systems, self-contained units, mechanical application—manual, and mechanical application—automatic.

Comparatively little has been done in improving the efficiency of roll neck lubrication—cold or hot necks—since the first mills were erected. Carload after carload of so-called tallow and hot neck compounds, have been dumped into the pits via the slab and stick route on to and off of the mill necks. A condition brought about largely because of the wasteful method of applying, because of little correlated action by machinery designers, mill operators and lubricant manufacturers to better the condition. Not only has there been the waste of lubricating materials, and excessive bearing replacement costs, but a waste of power undoubtedly at a cost far in excess of the other factors of lubrication costs.

Some conscientious mill lubrication engineers have made an earnest attempt to correct this. Ofttimes with little encouragement or support from the millwrights and greasers. Generally with the handicap of being unable to obtain the right type of grease for reasons best known only to the higher executive officials who cannot know the extent they are obstructing improvements in operating practice, their earnings, and encouraging a "don't give a hang" attitude on the part of the mill men, or they would not do it. Generally speaking, roll neck lubrication has shown little improvement in over 200 years or more.

Fig. 1 is an illustration taken from a book called "De Ferro" or "Of Iron" published in 1734 by Emanuel Swedenburg, and written in Latin, shows a characteristic "Iron Works" of the first half of the 18th Century.

You will note in the lower left hand corner what appears to be an old "Bloomery" in which ore is reduced by charcoal to a shapeless pasty lump of wrought iron. Near at hand is the smith who is reducing this shapeless lump to a more or less rectangular "bloom" suitable for inserting between the plate rolls. Above the bloomery is shown the reheating furnace for the blooms.

In the center of the picture is shown the two stands of rolls driven from the two water wheels, the two rolls being driven from opposite ends, a practice which is just about to come again into practical use.

The stand of rolls on the left are the plate rolls, between which the hammered bloom is passed repeatedly with several reheatings, until it is reduced to the thickness desired. Note here the trough from the head race of the water wheels to supply water to the rolls.

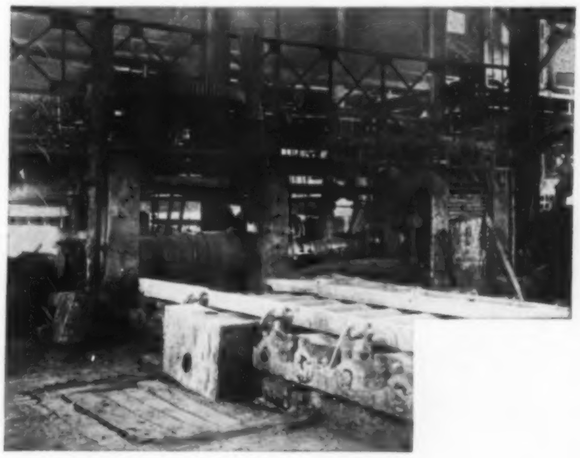


FIG. 2.

The stand of rolls to the right is the slitting mill which consists of overlapping discs, exactly like a modern slitter, between which the hot plate is passed and divided into a number of rods or bars suitable for further manufacture. It was from rods made in this type of slitting mill that the old fashioned wrought iron nails were made. The "Nailers" taking home a bundle of rods and working them up in a home smithy, into nails. Small wonder that nails were of great value in those days and even served for currency.

Fig. 2 is of a Blooming Mill built in 1877. From standpoint of lubricating, it differs little from the mill built in 1734. Mills of the open neck type providing for the same means of lubricating have been built within the last year or two.

Figs. 3 and 4 are two different views of another old timer, showing the open pinions.

Fig. 5 is a 3-High Bar Mill built in the early 1900s.

Fig. 6 is a 128" Plate Mill.

Figs. 7 and 7A show a modern 3-High Pinion and Reduction Gear. These two units were built by the builders of the old type of mills shown. (Mackintosh-Hemphill Co.)

The mills shown in Figs. 2, 3, 4, 5 and 6 compared with these modern machines clearly show the great strides made in machinery design and construction, generally speaking, but there does not appear to have been any great improvement made in the means of lubricating them.

It was noted in referring to a Blooming Mill built in 1895, that the roll chucks of these old mills did not differ materially from the more recent mills where paddle greasing is employed. (Fig. 8).

Some of our older members can undoubtedly recall such mills as these. Some of the younger members, who do not get away from their plants very often, may find them interesting and probably will appreciate better the superior conditions under which they work.

Smelly, wormy suet is still being used today as shown by questionnaires on roll neck lubrication received during the preparation of this paper. You do not have to put up with this. The situation even with the most obsolete type of equipment can be bettered, the cost of which should be quickly absorbed by the reduced maintenance, power and lubricant costs. Your lubrication engineer can do it, but he must be given the proper authority and have the moral support of Pete, Mike and Jack in the mill, the engineering department in helping him to work out his ideas, and the mill officials by giving him a free hand in the selection of lubricants. This cannot only be done, but it has been and is being done. There are two or three men in this audience that can vouch for these statements. They were a lot longer about it than they should have been, but it was not their fault.

Mr. W. A. James of the Bethlehem Steel Company, Lackawanna, N. Y. in his paper presented before the American Iron & Steel Institute in New York on May 9, 1930 stated:

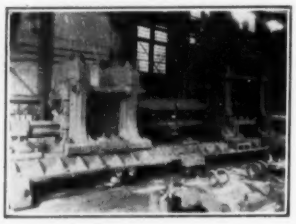


FIG. 3.

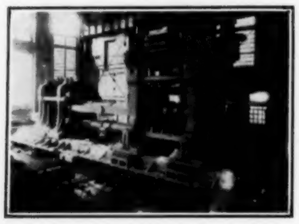


FIG. 4.

> "Selecting the Correct Lubricant. Mention has already been made of the correct lubricant to be used in certain specific operations and instances. The importance of exercising care and discrimination in the choice of lubricants cannot be over-emphasized. Such care and discrimination are in some ways becoming more, rather than less, difficult to exercise."

Mr. James' point is well stated, but we believe that if lubrication engineers were permitted greater freedom in the selection of their lubricants, without interference, and were not limited in the sources of their supply, they would not find it so difficult to overcome some of their problems.

These statements refer primarily to the lubrication of blooming and other types of reversing mills. Much progress has been made in applying roller bearings to other types of mills, but as usual, lubrication has been batted around with every change of the wind. There is not enough joint co-operation by the various ones interested—too much single handed pioneering—therefore too many "half-baked" conclusions.

TYPICAL THREE-HIGH BAR MILL

FIG. 5.

FIG. 6.

The qualities and characteristics of mineral oils have a direct influence on the lubricant in which they are used, therefore on the lubrication of roll necks. As most of us know, there are quite a variety of distinctly different types of crude oils. In addition to the Pennsylvania, Mid-Continent and Texas, are the crudes that come from California, Wyoming, Smackover and other districts. These are only a few of the many. Basically, however, for our consideration, we have four distinct groups.

1. The pure paraffin base. From Pennsylvania and W. Va. chiefly.
2. Mixed base crudes. From several parts of the country. They may be asphalt and paraffin, or naphthene and paraffin, or asphalt, naphthene and paraffin bases.
3. Pure asphalt crudes. Coming from several districts, including Mexico and Southern California.
4. Texas Oils. Usually known as asphalt or naphthene base oils.

Therefore, the values of heavy duty mill lubricants cannot be properly judged without a thorough knowledge and analysis of the Mineral Oils used in their manufacture. A lubricant is only as good as the oil used in its manufacture, is the common statement of authorities. The characteristics of all materials used in the manufacture of the lubricant, and the method of processing, should be carefully considered, as the several factors and processing relate to the work to be done. The usual factors of gravity, fire, flash, viscosity, pour test, etc., are indispensable when determining the type of oil and in checking values of different shipments of the same type of oil. Viscosity has been looked upon to a very large extent, as the chief means of measuring the values of an oil, but this factor can be misleading when used alone in determining values and suitability. These factors are also useful in considering the value of a lubricant for a specific purpose, but the particular characteristics of the mineral oil must be kept in mind in connection therewith. The types and qualities of the fats or chemicals used are also important.

A few years ago we were but little interested in fluid lubricants for roll neck lubrication. Practically all types of mills are now employing free flowing oils to a greater or lesser degree, including blooming mills, therefore it seems fit to discuss these types of lubricants.

At the present time there is a vast amount of research work and study being conducted to develop means of increasing the film strength or load carrying capacity of compounded lubricants—oils and greases. Their development was through the needs of gears, but their use has been extended to roll necks. Lead oleates have been used for this purpose for many years, first I believe by the railroads for car journal bearings, and later by other industries. The past two or three years unusual attention has been given to the use of lead compounds.

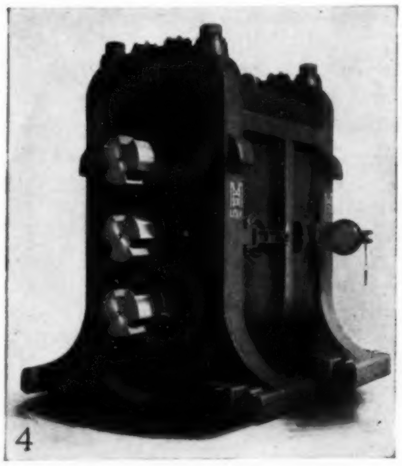


FIG. 7.

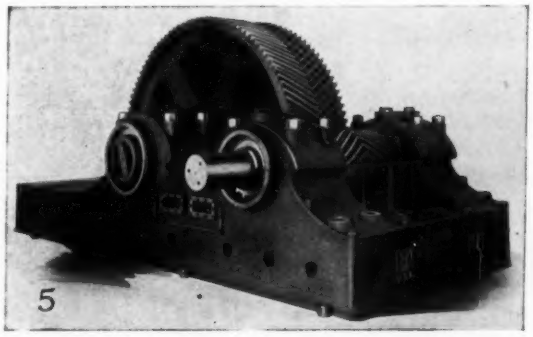


FIG. 7A.

Changes in design of geared units; closer tolerances, greater torques, heavier tooth pressures for steel mill equipment, as well as automotive transmission and differential units, has created a demand for a new type of lubricant that will more nearly meet these demands than those lubricants heretofore manufactured. The research laboratories of the larger industrial organizations, manufacturers of automobiles, manufacturers of roller bearings, machinery manufacturers, and practically every organization of any importance in the oil industry, have been and are devoting considerable time to the development of these new lubricants.

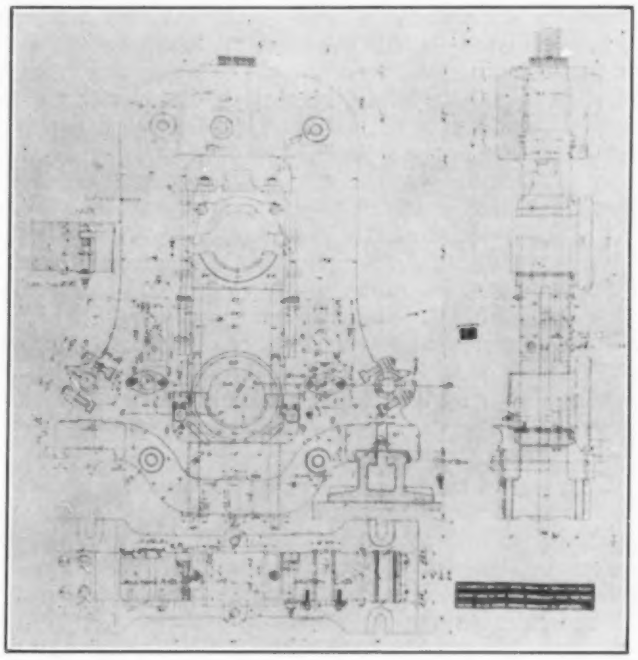


FIG. 8.

Insoluble lime soap base greases, ranging from a heavy tallow compound to be cut up into slabs, to the softer types of cup greases, are most commonly used for mill purposes. They are of a colloidal structure and contain more or less moisture; the better qualities a very small percentage. When the moisture evaporates, which may be caused by atmospheric or operating conditions, including line friction of lubricating systems, the soap and mineral oil may separate, decreasing the lubricating value of the grease, if not completely destroying it. Even the vibration of a machine will cause separation of the base from the oil used in a grease cup. The false body of the colloidal structure built into grease and heavy oils by various processes, has its useful purpose primarily in assisting in preventing the lubricant working out of openings too freely.

Well posted grease salesmen, knowing that the colloidal structure of grease is broken down by stirring it, always use fresh samples to show prospective customers.

Extreme pressure greases are generally lime base greases to which have been added lead oleate, sulphur chloride, or other film strength building qualities. They are made in various grades, determined by the grade of oil used, and in differing consistencies depending upon the amount of base used. Most of these greases, while technically known as insoluble, take up water more or less readily, and are washed away comparatively freely if too much water is used. This is particularly true of the softer consistencies.

A recent development for general mill lubrication is a line of anhydrous lime base greases. They are absolutely free of moisture. They can be heated up to the flash point of the mineral oil, when it becomes definitely a fluid, heavy or light according to the mineral oil used, but on cooling it will go back to its original consistency. In fact, becomes slightly heavier as when heated to the point the oil will flash, some of the lighter fractions are distilled off. This product is highly resistant to water. It can be applied on a polished surface under a flood of water. It will not wash off when flooded with water under hydraulic pressures commonly used in mills. When kneaded, such as with a spatula on a plate, or if there is a kneading action between the bearing and the neck, water will mix into it in the form of an emulsion and in a considerable amount, without detrimentally effecting its lubricating qualities. If such water added to the grease should become evaporated, the grease goes back to its original consistency, in the same manner as when tested under a high heat. It will not break down or separate.

This new anhydrous grease is made for heavy duty roll necks with varying degrees of tallow in the base, ranging from 10 to 25%, and a 600 fire test steam refined cylinder stock. Nominal amounts of finely grained flake graphite are used. For roller bearings, sleeve types of bearings and other similar purposes, it is made with varying grades and types of oils, from the light paraffins 100 @ 100 to the heavy vacuum distilled products. These grades are also made with varying degrees of base. All of them in varying degrees of consistency, the fluidity depending upon the type of mineral oil used, but all with the same per cent of base or tallow.

A special advantage of this anhydrous grease is that fluid numbers may be had for those purposes where heavy oils are now used. As it contains no moisture, it will not separate or break down under pressures within its limits. This type of product is not made in the form of heavy compounds, to be cut up in slabs and used on open necks. It is especially useful for mechanical lubricating systems.

When required, these anhydrous lime base greases will be compounded for extreme pressure uses. The ordinary type being known as Metallic-Grease; the extreme pressure as E. P. Metallic-Grease.

The new Hot Neck Greases are sodium base greases with a high melting point—in excess of the melting points of the stick lubricant commonly used on hot necks—and made in suitable plasticities for application with mechanical lubricating systems. This is a special product developed by us. Hot neck lubrication costs—power, bearings and lubricants—are very materially decreased by its use. The base is made of a high grade tallow, in varying percentages according to the requirements, and a high fire test steam refined cylinder stock. There is some flake graphite used.

The statement has been made in a report by the Carnegie Institute of Technology that from 60% to 80% of the power required in rolling sheets is used in overcoming roll neck friction.

Mr. B. S. Burrell of the Inland Steel Company, stated in his discussion on "Steel Plant Lubrication" before the American Iron & Steel Institute meeting, May 9, 1930, the following:

> "Sheet Mill Roll Train lubrication is the most expensive work in this line today, and it is hoped that a remedy can be found to cut down this extremely heavy friction loss and wear on the roll bearings. A statement for year 1929 of the cost of brass bearings and their lubrication in our Indiana Harbor Sheet Mill Department composed of 18 Mills, is as follows:
>
> Brass bearing cost per net ton of sheets produced—.085c.
>
> "Lubrication cost per net ton of sheets produced—.085c."

Still another authority has stated that a 10% reduction in hot neck lubrication—including power, bearings and lubricant costs—would result in savings that no mill executive could afford to ignore. By the use of this new Hot Neck Grease, mechanically applied, savings in excess of 10% may be expected.

As the major portion of mill equipment has the standard type of babbitt and bronze bearings, it would probably not be amiss to quote from an article, discussing a group of papers read before the Iron & Steel Division of the A. S. M. E. last Fall, in the March 25 issue of the ENGINEER published in London, in which it is stated:

> "It will be noticed that in the case of each of the types of bearings discussed at this meeting, running tests were claimed to show results superior to those from plain metal bearings. In view of this actual or potential competition from so many directions, it is interesting to consider what advantages the plain bronze and white metalled bearing can offer at the present time, and what are the prospects of improvement in their performance. In the first place, the plain bearing has in its favour a low initial cost, and it is difficult to see how this advantage is likely to be lost to any of its competitors, at least in the larger sizes. Plain bearings can be replaced quickly and cheaply, and in making the replacement "brasses" no special plant is required, other than that found in almost every foundry. The bearings have a good scrap value, and when they are worn down can be remelted to make new bearings. With effective lubrication and correct proportions, plain bearings, made from suitable metal and cast with reasonable care, have long given satisfactory service, except in conditions which are extremely arduous as regards speed, load, and temperature of the bearing surfaces.
>
> "The trend of modern engineering design towards higher speeds and greater compactness leaves little doubt that what are considered to be extreme conditions today will soon be regarded as normal and commonplace. This statement is as true of bearings as of any other component of engineering equipment, and it has a special significance for the manufacturer of plain bronze and white-metalled bearings, since it means that until he can effect radical improvements in his products he may anticipate not only a shrinking market for them, but also an increasing invasion into the field of general bearings by ma-

terials and devices, typified by the examples described above, which have been evolved primarily to meet the requirements of some specially severe type of service.

"It is now almost a hundred years since Babbit metal was introduced, sixty years since the 'invention' of the bronze with 10 per cent of tin and 10 per cent of lead, and over thirty years since patents were granted for the use of bronzes containing over 20 per cent of lead. When it is considered that these are still essentially the standard alloys in use today for the vast majority of plain bearings, there is not much evidence of progress in this particular field of metallurgy, and indeed reference to the excellent bibliography in Corse's 'Bearing Metals and Bearings,' shows how little fundamental research has been carried out on materials suitable for bearings. The number of secret mixtures and nostrums for bearing alloys still offered for sale every week is another eloquent commentary on the state of knowledge of this subject. There are, however, encouraging signs that the rate of progress is very definitely quickening. The British Non-Ferrous Metals Research Association in this country and such bodies as the American Society of Testing Materials in America, are giving the matter their serious attention. There is considerable leeway to be made up, but there is good reason to hope that these researches will lead to a new conception of the possibilities of plain metallic bearings, both in improvements in materials already in use and in the application of new alloys."

We will now discuss the different types of bearings, their design, lubricants, method of applying, etc., as outlined in the first part of this paper.

Standard Babbitt and Bronze Bearings (a) Open Neck Type.

These are so common, we question if there is a mill in the country that does not have equipment of this type, so it would seem useless to discuss their design. However, there are modifications of design that can be made to very good advantage. I am still continuing to preach properly located, more carefully cut oil grooves, and chamferred with greater care. In our preparation of this paper it has been emphasized a number of times the superior results to be obtained by finishing the bearings properly and polishing the necks. We have a Plate Mill which we serve, which the owners say is the toughest lubrication job in the country. Bearing costs have been very excessive in spite of using the best types of greases applied by a mechanical lubricating system. The bearings have been machined and finished in the usual manner, which is almost not at all, and the necks were not polished. In recent months the bearings have been accurately finished and the necks polished, resulting in a very decided reduction of bearing costs.

We earnestly recommend your consideration of the elimination of wearing strips or bronze gibs as an important means of reducing costs of babbitted bearings. Mr. P. Ruff who has been associated with us for some years, and who has had many years practical experience as a babbitt man, has been advocating this for many years. I recall distinctly in a conference some two or three years ago in the office of one of our leading machinery builders, in connection with a 40" Blooming Mill for which automatic lubrication was being considered, Mr. Ruff asked that the wearing strips be left out, but they had already been put in so the mill was started up that way. In due time, the mill was induced when repouring the bearings to leave the wearing strips out, which resulted in a very material increase in tonnage; namely, from 20,000 tons with the wearing strips to 125,000 tons without them. The gain would have been even greater had the bearings, which were 24"x36", been provided with two points of lubrication instead of one, and that one also had to take care of the thrust. Mr. Ruff further states in effect:

> "When using wearing strips, which almost invariably results in high friction and scoring of the roll necks, it is foolish to use good metal, as with the wearing strips the babbitt is wiped out and a cheap metal would answer under the circumstances just as well. There are generally three wearing strips inserted. If one cannot see his way clear to take them out entirely, take out the bottom strip and try it. This always results in a greater tonnage on a set of bearings."

If Mr. Ruff's position needed any support, he would have it in an abundant measure in the following report accompanied by tabulations from one of our leading Western plants.

"Lubrication Study Report."

> Some years ago, believing that we were not obtaining low enough lubrication costs, bearing practice was a constant bone of contention. The angles of discussion are known to you, therefore need no repetition. Ultimately a plan was adopted covering a thorough study. The mill stand bearings, being the most important from a cost standpoint, were chosen as the field of attack. Our bearings in general were babbitted with brass thrust collars and wearing strips, a practice handed down from year to year. Our first observation was that following their period of service the bearings were invariably burned out, especially adjacent to the thrust collars and the babbitt between the wearing strips. The question to decide, was this due to the babbitt, the lubricant, or was it a condition that had to be put up with because of its being impossible to overcome.
>
> As matters stood, when a bearing gave out, the operating men as they always have done, condemned the lubricant or the babbitt, or both. We were not satisfied to let the matter ride, so we started some tests on just one set of bearings on one of our larger mills. Our first attack was the method of pouring and finishing the metal. Various kinds were tried. Our answer was not there.
>
> Next a study of the lubricants and the method of applying was undertaken, force-feed and paddle packed. Several months were required to follow these tests through, which

were made on a sufficient number of bearings to determine that it was a standard condition. Of course improvements were made, but nothing to get excited about.

Our next move was to eliminate the brass thrusts and wearing strips. That was our answer, although we did not know it at the time. The first bearing tested resulted in an improvement of four to five times the tonnage over any previous experience. Frankly, this was unbelievable, and was given little consideration until further tests were made. The more tests we made, regardless of any other factors, the more certain was the fact established that wearing strips were harmful rather than helpful. Further tests established that the heavier thrusts required brass. Brass grids were used effectively, and finally adopted where needed.

We were now ready to establish a better practice, including the metals and their design; the lubricants and their application. Our study was carried from one mill to the other and always with the same convincing answer, leading to the final following conclusions.

(a) Paddle packed necks. The bearings to be supplied with thrusts of brass, and the body to be of babbitt with suitable grids where required.

(b) The lubricant to be of uniform consistency and homogeneous. The necks to be swabbed before and after running when necessary, and to be properly taken care of during operation, so that when roll changes are made the grease film thereon must indicate adequate service.

Experiments with force-feed systems, eliminated the human element to a considerable extent, and produced much better results, not only in bearing life, but a saving in power, less water used, and the sections were held more easily. The following table shows the increase in bearing tonnage resulting from this study and the changes made.

Increase in Bearing Life,
Based on Tonnages Rolled by Change
in Bearing Design and Better
Lubricant.

	1929	1930	1931
#1 Mill	52% over 1928	38% over 1929	
#2 Mill	31% over 1928	3% less 1929	
#3 Mill	31% less 1928	69½% over 1929 15% over 1928	20% over 1930

I presume some will persist in the use of slabs of grease for the lubrication of standard babbitt and bronze bearings, but it will require but very little ingenuity to provide for much more satisfactory lubrication by using more suitable types of lubricants. One of our mills, through the cooperation of an Eastern grease manufacturer, has been using a rather soft grease in a three compartment box, the soft grease being placed in the inner compartment and hard grease in the outer compartments, acting as a wiper to the soft grease, and as a seal against wastefulness. I am not in the position to state the figures, but I know that bearing life has been very materially increased and lubrication costs greatly reduced. We have been lubricating open neck types of bearings of the crudest design for three or four years with a mechanical lubricating system, and it has been exceedingly successful in every way. It may not always be easy to adapt old equipment to mechanical lubrication, but the certain gains make it well worthwhile. This is particularly true where there is difficulty in holding the thrusts on the bearings. I know of no better way to cover this than to quote data supplied by Mr. E. J. Thomas of the Republic Steel Corporation of Buffalo, N. Y., which is as follows:

"1. **Hand packing with hard roll neck greases.**

"As made today, these greases provide an excellent film of lubricant for the roll neck so long as the grease can be kept in direct contact with the roll neck. Unfortunately, it is only by very frequent packing and repacking of the grease in its pocket or side space, that this contact can be maintained. This is the exception, rather than the rule, with the result that on mills where this type of lubrication is relied upon, an effective film of lubricant is maintained on the roll necks for only a very small percentage of the time, and approximately 80% of the grease is wasted. The wear on the bearings, then, is accelerated by the metal to metal contact between bearing and roll neck, which inevitably occurs. Also power waste.

"On large mills with large bearing surfaces, this wear does not seem so noticeable until effective lubrication is introduced through the use of other methods, but on smaller, higher speed mills the number of bearings changed due to inadequate lubrication is startling.

"On the smaller mills where various shapes are rolled, and where close tolerances are demanded, the portion of the bearing which wears the most is the collar. That is perhaps due to the high pressures imposed upon this surface due to the lateral adjustments made in getting and maintaining the proper section, and also because the area of this bearing surface is so much smaller than the horizontal or neck area. In addition, this collar surface is also found to be the most difficult to keep supplied with lubricant in contact.

"The collars of these bearings are usually of solid bronze, and on the necks large babbitt inserts are provided for in the bronze grid. Nevertheless, the wear on the collars will often amount to eight times that on the necks.

"The problem then, is to provide a means of lubrication which will keep an effective film of lubricant first on the collars and also on the necks of the roll neck bearings.

"2. **Pressure Grease Systems.**

"The main purpose of these is to supply grease under pressure to the roll neck bearings in amounts that may be adjusted to just what is needed, and as often as may be found to be desirable. The first of these to come into use were hand operated and still required the attendance of at least one man to insure its effective operation. They are now automatic in their operation, and are made to operate only in coincidence with the mill. There are at present a large number of systems on the market, all with various advantages and disadvantages, but it is easy to select one which will fit into the type of mill to be lubricated with the minimum amount of cost, upkeep and room required. Most of these can deliver the right kind of grease to the bearing in the proper amounts and at the proper times if correctly installed.

"The main question is what to do with the lubricant when you get it to the bearing and also to make sure that you get the best type of lubricant for the particular application. This is not so simple as might be supposed, and experiments are still being carried on to determine the best design for roll neck bearings for different types of mills, and as yet there are few accepted tests for determining the proper type of lubricant other than using the lubricant under actual operating conditions, and this requires considerable time. There are various ideas in circulation as to what ingredients and proportions of same an ideal roll neck grease should contain, but it is certain that the grease must be able to maintain a good lubricating film (low friction coefficient) under conditions of great pressure and fairly high temperatures, and resist the washing action of generous quantities of water.

"We have at present two pressure grease systems in operation, providing lubrication in two different mills. One is operated by hand while the other is run by electric power and is tied up with the mill motor so that it operates only when that motor is turning over.

"The hand operated outfit is used on a 14" stand which does not run at a very high speed, but which is subjected to intense and rapidly changing pressures. Prior to the use of this system difficulty was experienced in maintaining bearings of any type in use for any length of time, and in keeping the proper tolerances. With the pressure greasing system in operation and attended properly, plain babbitt bearings with grease grooves have proved adequate and give excellent wear.

"The System operated on electric power is at present lubricating the finishing stand on a 10" Mill, which is run at high speeds and experiences considerable side thrust on its bearings. The System is now considered essential to maintain the bearings and prevent their very rapid wear. To date, however, the increase in the quality of lubrication has not changed the ratio of wear on the collars to that on the necks. The horizontal surfaces of the bearings have only just started to wear when the collars are worn out. This presents the problem of maintaining a more effective film on the collar, and this is being given considerable attention.

"The grease used in most pressure systems is necessarily softer, more plastic and of a lighter consistency than the hand-packed hard roll neck greases. This is thought to reduce the internal friction of the lubricants used, but there is some question as to whether the pressure resisting qualities of lighter lubricants are pronounced enough. Our experience is not yet sufficient to indicate definitely whether or not this is so, but the lighter greases now in use in our pressure systems are giving good service.

"My opinion, from such experience as we have had, is that pressure greasing systems can be installed to advantage in practically every type of mill, except perhaps, the smallest and fastest ones. The possibility is that small mills originally designed for forced lubrication would show great savings in bearing wear.

"Contrary to general opinion, the consumption of grease does not increase with the installation of automatic greasing systems. If the proper amount of grease required by the bearing is once determined and can be supplied readily, it is quite possible that large savings in grease consumption can be effected."

As you will have noted, Mr. Thomas refers particularly to the difficulty of providing proper lubrication for the collars or thrusts. He states that with a lubricating system the ratio of wear is not materially bettered. We suggested that he might be able to better conditions by applying the lubricant to the thrust at more frequent periods mechanically. Mr. Thomas' reply to that in part is as follows:

"We are at present attempting to work out a method of applying the grease direct to the thrust collars of the bearings and arranging for a large part of the grease to work from that point back across the neck surface of the bearing through suitable grooving, aided by whatever spring the bearing may experience.

"The great difficulty in this particular instance is a lack of available space which makes it inconvenient to establish multiple grease leads to various points on the bearing. Because of this lack of space, we are working with one lead. As you doubtless understand, these 10" stands are not very large, making it desirable from many points of view to simplify the application as much as possible.

"I agree that separate and more frequent applications direct to the bearing thrust collars would undoubtedly improve the wear of the entire bearing. Indeed, so marked is the wear on these collars in contrast to the neck surfaces that it is believed a great increase in life would result if only the collar were to be properly lubricated, with no provision made for the neck surfaces. However, it is readily seen that power and frictional losses make proper lubrication of the necks very desirable, although in my opinion, of secondary importance to perfect lubrication of the thrust collars. This holds, primarily for small high speed mills where the wear on thrust collars

> is pronounced, and may not be so important for larger, slower mills.
>
> "If two or three leads, one direct to the thrust collar, were feasible for each bearing it would be possible and just as satisfactory, I believe, to deliver different quantities of grease to each bearing surface—the greatest quantity at the thrust collar, and at very frequent intervals. The period of delivery could be the same for each surface, but the varying amounts could be so regulated that the net result would be the same as varying the periods of delivery to each bearing surface, as you have suggested. This is possible with several systems, impossible with others. I believe both methods would be possible with the Adams-Schuette System, and I agree that your point is a very pertinent one, where it is convenient to use more than one grease lead to the bearing. I also believe that in designing new mills and new bearings for such mills, provision could be made for at least two leads to each bearing, even for the smallest mills."

We believe that Mr. Thomas' thoughts on this subject are worthy of careful consideration, especially in connection with thrust lubrication.

We consider the independent and direct lubrication of the thrusts of the utmost importance, whether oil or grease is used. Means should be provided for close adjustment of the amount of lubricant applied, and the periods of application should be subject to close regulation, and wholly independently. With an over-supply of grease to the thrust, which would be the result of using a common set period of application, because of the play usually between the bearing and the roll fillet, resulting in a more or less open space, there must be considerable waste. We emphasize the advisability of applying the grease to the thrust in very small quantities at frequent periods. By this procedure, even though when at the time the grease is discharged to the thrust the parts are separated, because of the small amount of grease delivered, if it has the proper adhesiveness, it will not drop or throw off as a larger amount would. Furthermore, with the frequent applications as the roll turns, the grease is applied to a greater number of points than would be possible with less frequent applications of a larger quantity of grease, therefore greater assurance of the proper lubrication. With automatic application a drop or two of grease can be applied every few seconds, thereby leaving no doubt as to there being lubrication for the thrust.

One mill operator makes this statement:

> "One item of interest is that on our 28" Mill installed in January, 1931, the bronze thrust collars gave very poor service, no matter what type of lubrication was employed. We substituted babbitt face collars with oil drip lubrication from tanks on the housing, and this arrangement has proven satisfactory."

This, we believe, to be because he is keeping a constant supply of oil between the fillet of the roll and the collar of the bearing. An extreme pressure type of oil might give even better results.

Fig. 9 shows a scheme developed by H. Slattengren for lubricating mill necks with oil, including the thrust collars.

I do not know of any way to go about proving it, but I believe if unable to obtain appropriations for roller, or other special bearing equipped mills, or where it is not practical to apply them, the proper effort in reducing friction loads by more carefully prepared bearings, the use of more suitable lubricants, properly applied, should result in power savings, of very interesting figures. The roller bearing authorities state that the most important factor under consideration by mill operators is the power saving. That is one of the main arguments offered for roller bearings.

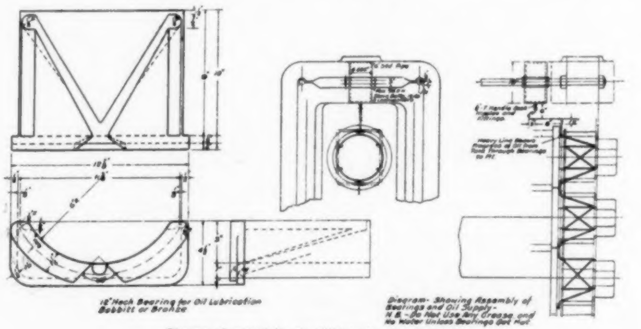

12" Neck Bearing for Oil Lubrication
Babbitt or Bronze
Diagram- Showing Assembly of Bearings and Oil Supply-
N.B.-Do Not Use Any Grease and No Water Unless Bearings Get Hot

Diagram of method for the oil-lubrication of roll necks

FIG. 9.

Standard Babbitt and Bronze Bearings (b) Sleeve Type.

There is considerable thought being given to roll necks with complete sleeve bearings. Mr. A. D. Adams of Spang, Chalfant & Co. has devoted considerable time to the study of this type of bearing, and his experiments which he started in 1924 have brought increasingly good results. On a 28" Sizing Mill started in February, 1927, the bearings of which are bronze, nearly 600,000 tons have been rolled on the original bearings, and they are still operating. Undoubtedly much of this success was due to the lubricating system used in this mill. It was operated manually until November, 1928, when it was made automatic as it is today. Continuous sleeve babbitted bearings are being designed for several units in this mill, including a 32" Blooming Mill. Mr. Adams states that this class of bearing can be operated in many places without water; takes less power and lubricants, and where babbitt alone is used higher polish on the roll necks is effected. (Fig. 10).

Another of our prominent mill officials has developed and has had in service for a number of years in an experimental way, a sleeve type of bearing, details of which we are not at liberty to discuss, which is proving very efficient. The ultimate results are exceedingly promising.

In recent months one of our machinery builders has illustrated and offered a sleeve type of bearing, for which they claim very low coefficient of friction. The oil recommended by the makers of this bearing is the ordinary oil used in circulating systems. If extreme pressure lubricants have the merit that our leading engineers think they have, and which is proved to be so by the results obtained in actual use, extreme pressure lubricants for this type of bearing should be more suitable than the usual type of circulating system oils.

Roller Bearings.

The response to our request for information for this section of our paper has been very good so far as the spirit is concerned, but any real operating data has been noteworthy for its absence. Mill owners have spent many thousands of dollars for equipping mills with roller bearings. They have had their joys and they have had their griefs. Designs of roller bearings are many. For their merits we will have to refer you to their makers.

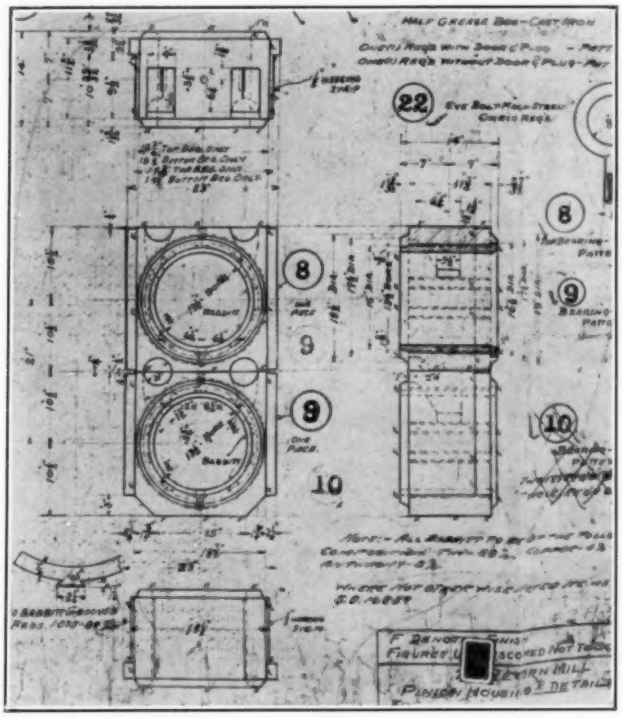


FIG. 10.

The differences of opinion regarding what are the most suitable lubricants and the best means of applying are about as far apart as it is possible for them to be. It has not been so long ago that some of the roller bearing manufacturers took the position that their bearings did not need lubrication except to keep them from rusting and to wash out foreign substances. In our judgment, not enough thought has been given to the lubrication of roller bearings. Oil was the preference to start with, but there was difficulty in providing the proper closures. Then grease became preferred, particularly for heavy duty units, and the roller bearing manufacturers not being able to obtain dependable information from the lubricant interests, have spent much time and money in their efforts to find suitable greases and means for determining their suitability. The roller bearing manufacturers responsible for this are entitled to a lot of credit, but they are still guessing. Some advocate the application of lubricant to the top of the bearings, others design the units to apply the lubricant at the bottom of the bearing. In one case the lubricant must pass on through the unit by gravity, and requires one kind of lubricant. In the other the unit must be lubricated by adhesion, which requires a wholly different type of lubricant on identically the same design of bearing. Which is right?

This is not in criticism of the anti-friction bearing manufacturers. They have to "cut and fit" until the problem is solved. I do feel, however, that they are not giving lubrication as much attention as they should, as much attention as they do give it. There is something lacking some place. The lubricant manufacturers may not be digging hard enough, or the roller bearing manufacturers may not be utilizing all of the sources of information that are available to them, or—and, I think this has a lot to do with it—neither the lubricant manufacturer or the roller bearing manufacturer are taking full advantage of all the information to be obtained through the men in actual charge of the equipment.

We consider the lubrication of anti-friction bearings of far greater importance than is held by engineers and operating officials in general, especially in connection with some of the later projects where the relative loads and speeds are more severe than heretofore. We are completely out of sympathy with manual application of greases, no matter what the method for applying it may be. The automatic mechanical application in adjustable predetermined amounts, at regulatable predetermined periods, should be employed wherever the service is at all severe or constant. While there has been some excellent work performed by roller bearings, there are requirements which still call for experiments, and dealing with factors unknown. Automatic lubrication provides means for better meeting the unknown conditions of the experiment to be made.

We recommend the type of lubricant best suited for the particular roller bearing conditions, applying it in measured quantities, at the proper periods, thus keeping a light deposit of grease or oil within the bearings without wastefulness. The general practice seems to be to fill the roller bearings with a lighter grease than should be used for heavy loads, and letting it operate until the grease is worked out, when a fresh amount is applied. This practice results in a variable condition from beginning to end, first a feast and then a famine, and does not permit the use of the kind of grease that should be used for the heavier jobs.

There are differing opinions as to the best method of applying lubricants. The Timken Roller Bearing Co. on certain heavy duty units, Figs. 11 and 12, prefer to apply it at the bottom, letting the application be by adhesion. I may be mistaken, but I believe within the last six months I saw an illustration of these same bearings where the lubricant was applied at the top. Fig. 13 shows the application of the lubricant at the top of the bearing. This particular application has been used with very excellent success, the oil being handled through a circulating system.

There has been some interesting information resulting from preparation of our paper in regard to roller bearings. In the September, 1931, issue of MACHINERY Mr. R. E. Brunner, chief engineer, S.K.F. Industries, makes the following statements in his article "Lubrication of Ball and Roller Bearings":

> "The frictional resistance in a ball or roller bearing increases with the load and speed, and unless oil of the proper viscosity, or a grease

of the correct consistency is supplied in proper amounts, the increase in the normal operating temperature may be serious.

Under ordinary conditions the temperature of a bearing while running will be from 10 to 60 degrees F. above that of the room. If it exceeds 125 degrees F. ordinary greases will frequently prove unsatisfactory. They will tend to soften and flow continuously into the path of the rolling elements, causing a rise in the normal operating temperature due to the increased frictional resistance introduced. This may eventually result in the separation of the oil and soap base, with a complete loss of lubricating qualities. In some cases, greases developed for use at high temperatures may be employed. Care should be taken, however, to see that they meet all the requirements for adequate lubrication.

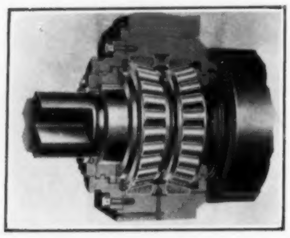


FIG. 11.

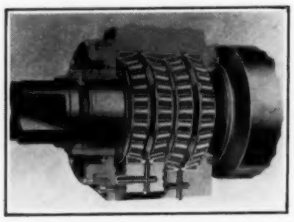


FIG. 12.

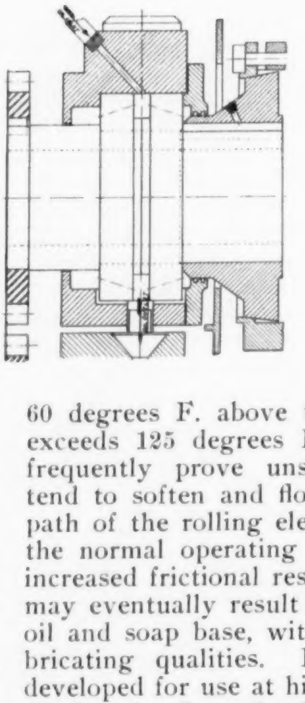


FIG. 13.

"In no case does a ball or roller bearing require a large quantity of lubricant. On the contrary, a few drops of oil or a corresponding amount of grease, properly distributed over the running surfaces of the bearing, will provide satisfactory lubrication for a considerable period of time. A large volume of lubricant within a bearing will usually result in high operating temperatures, due to the working or churning of the lubricant by the rolling elements and retainer. This may seriously impair the useful life of the lubricant through oxidation or sludging of the oil or actual disintegration of greases.

> "Grease is being used successfully for the lubrication of ball bearings at high speeds, but great care is necessary, both from the standpoint of housing design and selection of the lubricant, in order to obtain satisfactory results. Any system employed must be designed to feed only a limited amount of grease to the bearing.
>
> "Roller types of anti-friction bearings generally provide a guiding reaction to assure proper alignment of the rollers with the races. Usually this is obtained by contact between the end or head on the rollers and a suitable guiding flange on the inner race. Since this contact must involve some degree of sliding action, it is necessary to lubricate these surfaces, and provision must be made for free access of the lubricant. An oil film must be maintained to support the guiding pressure at the operating speeds and temperatures."

A letter of April 1 from Mr. Brunner states in part:

> "There is unfortunately a wide variation in opinion among authorities as to the correct lubrication of ball and roller bearings for rolling mill equipment."

Mr. E. C. Gainsborg, formerly of the S.K.F., now associated with the Roller Bearing Company of America, expresses a great personal interest in roll neck lubrication and states in part as follows:

> "The successful lubrication of a roll neck bearing is only one of three important fundamentals which must be considered when anti-friction bearings are to be used on roll necks. The first is an intelligent selection as to design and capacity of the bearing to suit the type of mill involved, and this is not restricted to size of stock which is being rolled, but rather to the type of mill, that is, if to be a two high, three high, four high or cluster mill.
>
> "The second point of importance involves the design of housing which takes in not only the importance insofar as the retaining of the lubricant and the exclusion of foreign matter, but also the ease of assembly.
>
> "The third is essentially the problem which you are concerned about, and that is, lubrication.
>
> "Unless the first two have been properly provided for, the matter of lubrication can be predicted to fail, and I say this with emphasis because I know of installations that have been fitted with proper lubrication, but because the selection of bearing and the design of housing has not been satisfactory, the service had in spite of a good lubrication system and proper type of lubricant, the endeavor has been a failure.
>
> "Quite aside as to what might have been the experience of others, my contact with the problem convinces me very definitely that for roll neck applications equipped with anti-friction bearings, oil must be used. Grease at its best is not satisfactory.
>
> "As this letter is dictated, there is called to mind a problem where the bearing was properly selected, the design of housing was as effective as could be desired and after the

mill was placed in operation, the lubrication failed. An attempt had been made to use sight feed lubrication, but it was not satisfactory. Considerable money was spent in installing a circulating system, and that, too, was unsatisfactory because water found its way within the housing. Finally, as a last resort, an emulsified cutting oil was used and this product has the property of readily absorbing water and through the use of that emulsified oil, it developed that as long as the proportion of oil in the water was in the amount of not less than 7½% that the bearings were lubricated satisfactorily and in that instance that procedure helped to solve the problem."

We replied to Mr. Gainsborg's letter above quoted in part as follows:

"We are particularly interested in your experience with emulsified cutting oils for roller bearing lubrication. The danger we see to this practice is because of the various types of soluble oils on the market in which all kinds of compounds are used, some of which I feel might prove corrosive. New compounded oils having water absorption qualities are now available which might be considered in this connection to good advantage.

"As to your statement: 'Grease at its best is not satisfactory.' We believe your position might be modified if you knew of the new type of anhydrous lime base grease which is now to be had. Containing no moisture, there is none to evaporate, therefore there is no separation from temperatures or extreme pressures."

In his letter of April 29 Mr. Gainsborg replies in regard to the emulsified cutting oil in part as follows:

"When I wrote you under date of April 12 and referred to certain experience when it was found necessary to use an emulsified cutting oil, I should have pointed out that the procedure was exercised as a means to an end. The results were more satisfactory than had been anticipated, and it occurred to me that this information, which I do not expect is generally known, might have been of some use to you. Personally, I feel that it is worth while looking into.

"The film strength of an oil is not so important a characteristic when considering it for use with an anti-friction bearing for the reason the load is not carried on the oil film which is the case with a plain or sleeve journal. The lubricant when used with ball or roller bearings serves primarily to protect the highly polished surfaces; also in certain types of application it functions in dissipating heat which might be generated due to an overloaded bearing, and there is still a third condition, and that is, that a bearing housing packed with grease may keep out the entrance of foreign matter into the housing. Of course, a lubricant also serves to reduce friction between contacted areas.

"I have said that oil is the proper lubricant for anti-friction bearings. When I made that statement I had in mind anti-friction bearings as used on roll necks or installations where the service is severe and where the unit is subjected to extreme changes in temperature.

"Let us assume that we are using a grease which is ordinarily selected to function around temperatures of 60 to 70° Fahrenheit in which range it will flow freely and should the temperature drop below that range and the grease be somewhat stiff, there is a tendency for the rolling elements of the bearing to channel out a path in the grease under which condition no lubricant flows between the rolling element and the roll path. On the other hand, assume that the grease has been selected for a 60 to 70° Fahrenheit range and the temperature should go beyond that to the range of 100° or thereabouts and the grease flows most freely, then you have the condition of it running out in the housing. A housing designed for use with grease makes no provision to take care of a freely flowing lubricant and if to provide for this emergency it becomes necessary to design an elaborate housing enclosure. The answer is evident: why use compounded grease when it is possible to use a straight mineral oil?

"For several years when associated with S.K.F. Industries, I had charge of that part of their engineering problem which tied up with lubrication and going back to those days it is my recollection that the difficult part of our problem was to obtain greases which would not separate out because of changes in temperature and further our problem was not made any easier because when we did locate several sources who manufactured a satisfactory grease, it just happened that they were smaller companies who did not have very large distribution, consequently, placing them on the S.K.F. approved list did not help the customers to the extent that was desired. You say that you now have a product which holds up from a range of below zero to 400 or 500° F. That information should certainly be of interest to bearing companies who are marketing their product for railway and similar applications."

Mr. Klinedinst of the Timken Roller Bearing Company emphasizes the power savings through equipping mills with roller bearings, stating:

"We ran a great many tests in our own mills here primarily to show power savings, they being the most important factor under consideration by the mill operator, and these tests have shown from 35 to 50% saving, and in some cases even as high as 65% saving, and we have been told further that without question on sheet mills of the 4-High type savings have been as high as 70% over the old 2-High type, but since most of the sheet mills equipped with anti-friction bearings are of the 4-High type and since none of these mills were ever equipped with plain bearings, there is no means of making any exact comparison. It may be possible that some of the mill operators will be able to give you data on plain bearing operation, but frankly in our

> four or more years of close contact with steel operators, we have never been able to get any factors of sufficient value to make a comparison between plain bearings and anti-friction bearings."

Among the most interesting letters we have received on this subject is from an official of an anti-friction bearing manufacturer, who states:

> "We know nearly nothing about lubrication. There are many engineers who use lubricants constantly but who are unable to predict what lubricant would be satisfactory for a new kind of bearing or a previously inexperienced method of operation.
>
> "We have been looking forward to you lubricating engineers to tell us how to proceed to measure the properties of your various lubricants; to explain to us what ingredients makes one lubricant more adaptable than another for a specific use. We can find no comprehensive literature extant on the subject. Lubricant manufacturers fear to tell what are the constituents in their lubricant. The result is that we have to experiment with numerous grades in order to reach a conclusion as to what brand ultimately to employ. It is for you to tell us what makes lubricants, what are lubricating properties and how to detect them. We are awaiting eagerly this presentation of your paper at the impending meeting of the A. I. & S. E. E."

We consider the statement I have just read one of the most scathing criticisms of the existing condition that could be made. Probably if a more earnest effort was made by anti-friction bearing manufacturers to learn the facts in regard to lubricants, the difficulty would be less. A closer co-operation between machinery designers and lubrication engineers—designing and operating with lubricating men in the field who are working with the most important factor of successful machine operation—its lubrication—will be mutually helpful to all concerned. As to the difficulty brought about by the lubricant manufacturers fearing to tell what the constituents are in their lubricants. We are unable to offer any suggestions until there is an enlarged spirit of fair dealing in the industry.

In most industries, if a manufacturer's organization develops a new product or method, whether he obtains a patent on it or not, others in the industry have some respect for his rights. That does not seem to be the case in the lubricating business. Let some manufacturer develop something wholly new, something better,—he may have spent months of time and a considerable amount of money in developing his new product—and the minute he offers it for sale there are other manufacturers of lubricants who obtain a sample as quickly as possible, send it to their laboratories and try to duplicate it, and then offer their imitation for something that is "just as good", generally at a lower price. What are you going to do with a situation like this: You tackle some difficult lubrication job, and the mill official tells you that even though you may overcome his difficulty, under certain arrangements with other manufacturers, they will have to let them have a sample of the successful product so they may duplicate it if possible, and buy from them thereafter, rather than from the manufacturer who developed it and got them out of their trouble?

Composition Bearings.

While some remarkable results from the use of this type of bearing have been reported, we do not find sufficient data to base any opinions on. Furthermore, papers will be read by several of the representative composition bearing manufacturers, which will probably cover all the information available at this time.

In our correspondence with the various manufacturers, and in studying their literature, we find some information which we will quote from. It is quite noticeable that each and everyone emphasizes the importance of polishing the rolls. Everyone has made a special point that the bearings will work satisfactorily with water as a lubricant, but they always add that the use of oil or grease will produce better results.

Mr. Arthur J. Schmitt of the Continental Diamond Fibre Co. read a paper before the American Society of Mechanical Engineers' Annual Meeting last Fall, containing much of interest. We quote from this paper:

> "The application of a superior lubricant such as oil or grease improves their performance appreciably. Celoron bearings serve most efficiently on highly polished roll necks and they have produced substantial power savings in comparison with metal bearings and their life in numerous installations has been found to be many times that of metal bearings.
>
> "In general, the significant facts are: First, that celoron bearings carry capacity loads with very moderate friction losses when water only is used as a lubricant; second, that the friction losses under capacity loads are minimized when oil is used as a lubricant; third, that with no lubrication whatever, capacity loads can be sustained for limited periods of time without destruction of the bearings."

In Mr. Schmitt's letter of April 18 he states in part:

> "With regard to lubrication of the phenolic type of bearings. We have no compiled data that we can present to you, and it will be some time before that information can be acquired.******** Some can and do tell us that where electric power is used, they find indicated savings of about 15%. The writer is personally familiar with one mill, which on account of being speeded up was greatly overloaded to the point where motor burn-outs were a serious problem. Celoron bearings so lightened up this load, they now have no electrical trouble whatever and their records are quite accurate, and show a power saving of 30%. Unfortunately for you, they use no lubricant at all except water, and while I have recommended that they do lubricate and secure additional savings, they feel that they have already accomplished much more than they hoped to, and therefore are quite satisfied to let things alone for the time being.
>
> "Tests in our laboratories show that a lubricated bearing is about 7% more efficient

from the reduction of friction viewpoint when compared with water lubricated bearing, and there is undoubtedly an additional length of life, but the rate of wear on these bearings is so light that we have no way of telling just how much longer a properly lubricated bearing will last, than one which is merely water lubricated."

Mr. H. M. Richardson of the General Electric Co., manufacturing the Textolite bearing, has offered some very good information in regard to operating data which is incorporated in the tabulation shown in lantern slides to be referred to later. In reference to preparing the necks he writes:

> "In all of the cases covered by our data, no attempt was made to grind the roll necks, before installing Textolite bearings. It was noted, however, in all cases, that after operating for a considerable length of time on Textolite, the necks were polished to a very smooth surface. If the necks are extremely rough to begin with, there is likely to be a certain amount of cutting on the bearing when it is initially installed, so would recommend that the roll necks be ground before installing Textolite bearings."

Mr. Richardson's company seems to be carrying on a very thorough study of the possibilities for this type of bearing, as well as an extensive research in connection with lubrication. Mr. Richardson writes as follows:

> "We are running, at the present time, a series of tests in our Research Laboratory at Schenectady, for the determination of the friction characteristics of the different grades of Textolite materials under pressures ranging from 500 to 2000 lbs. per square inch, and rubbing speeds from breakaway to 3000 feet per minute, with various types of lubrication, including,
>
> 1. Water only.
> 2. Combination of water and block grease.
> 3. Soft grease only.
> 4. Flooded oil lubrication.
>
> These tests are well under way, but final conclusions have not been reached.
>
> "As an indication, however, of the field of application for Textolite roll neck bearings, with water lubrication, or combination of water and block greases, we have found that the coefficient of friction ranges from .004 to .008, for pressures from 500 to 2000 lbs. per square inch, and rubbing speeds from 250 to 3000 feet per minute. Below 250 feet per minute the coefficient of friction rises very rapidly."

Fig. 14 and 14a shows the method of applying water to the Textolite bearings, and an illustration of the Textolite lining inlaid in common bearing brass.

Mr. K. T. MacGill of the Joseph T. Ryerson & Son Co., manufacturing the Glyco bearings, has furnished some operating data which is included in our tabulations.

Method of Applying Mechanical Lubrication and Preparation of Bearings. When discussing mechanical lubrication, particularly for old equipment, invariably the question is asked, how are we going to apply it? How should the bearings be prepared? One might as well understand at once that it requires study and work. Sometimes a lot of it. But, the efficiency of the mill will be very materially increased through lowered bearing costs and saving in power waste, at a cost of far less than the same gains could be accomplished by completely overhauling the mill or buying a new one.

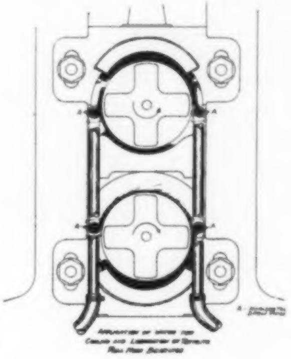


FIG. 14.

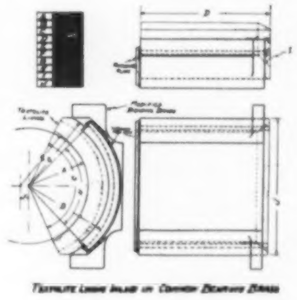


FIG. 14A.

There are so many conditions to meet that it would seem impractical in a presentation of this kind to make any definite suggestions. Get the grease to the bearing chuck or bearing in the easiest way your conditions permit. It can be done. It will pay.

Bearings. The preparation of the bearings for the proper distribution of the grease over the bearing surfaces is highly important. Greases will follow the lines of least resistance, the degree depending upon the type of grease—its plasticity, fluidity, and adhesiveness. If the pressure is not equal throughout the entire bearing area, the lubricant will work into the parts of lowest pressure more freely than it will into areas of higher pressure. Figuratively "robbing" some points and "flooding" others. It is frequently advisable to have two or more lines to each bearing. As stated before, adequate provision for the thrust collar is important.

The illustrations now shown are examples of bearings that have been prepared for mechanical application of lubricant. Fig. 15 shows the bearing of a 40" Mill with only one grease line, which is not enough for the best results for a bearing 36" long. There should also be a direct line to the thrust for this mill. Fig. 16 shows the bearing for a 24" Mill, where the grooves are very similar to those commonly used with oil. These are not large or deep enough; and in our judgment could be used only for oils and the lightest types of greases. Fig. 17 shows the old type of open neck bearing. This arrangement has been in use for three or four years with a semi-plastic lubricant. Results have well justified the trouble and the expense for preparing these bearings. Fig. 18 is an arrangement showing one line to the barrel of the bearing and one to the thrust with several outlets therefrom at different points of the thrust. It has been suggested that brass studs on the thrust at the points where the openings are made would be helpful. Fig. 19 shows an arrangement for distributing the lubricant over the entire length of the bearing. This method has been tried, but was not successful. This may have been because of not using the right type of lubricant. In any event, we believe a further effort should be made to develop this arrangement.

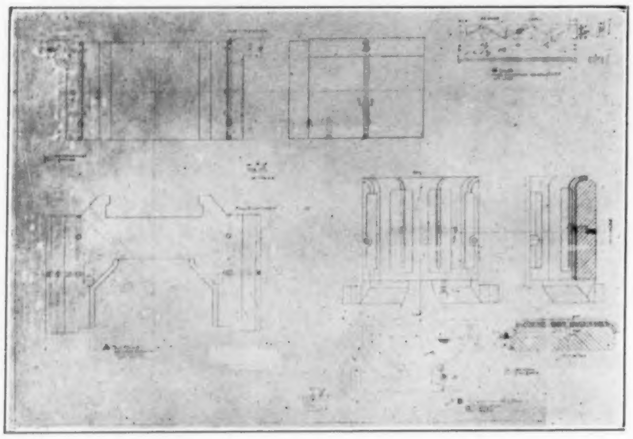


FIG. 15.

Referring to the various types of bearings and comparative data offered by the makers, one of our most active lubrication engineers states:

"There is little reliable data to be obtained on composition bearings. I find that nearly everyone guesses at the life of the plain bearings and exaggerates the comparison.

"This same holds true of roll neck roller bearings. They have never been tested against automatically lubricated plain bearings, and again the comparison is exaggerated.

"Some fairly reliable data may be obtained on plain bearings hand lubricated. These values are subject to considerable variation depending on material rolled, frequency of roll changes, variation of foundry practice and on the fickle interest of the greasers. Automatic lubrication smooths out the effect of these variations and results in greater bearing life."

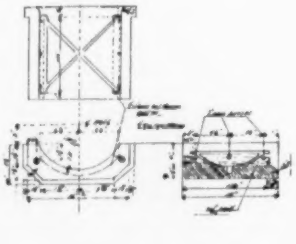


FIG. 16.

Our request for roll neck lubrication data to present to you was answered by most of the mills to the effect that they did not have the data required. However, a number, some at considerable trouble, sent the information they had which has been tabulated for each type of mill. Fig. 20 shows a tabulation of the blooming mill data. We refer you to the Association offices for the tabulations covering other types of mills.

While much has undoubtedly been accomplished, unquestionably there is a lot yet to be done to bring about economical lubrication of roll necks, and to be in step with the progress that has been made in general in machinery design in other ways.

Steel mills do not hold any patents on this situation. Mr. C. G. Williams, Consulting Mechanical Engineer, in a discussion of papers on roll neck bearings, delivered before the A. S. M. E., referring to paper mills stated:

> "This is a business where, admittedly, machinery and bearings are on the whole one hundred years behind present practice, and where pressures per square inch are but normal compared to pressures of rolling mill machinery."

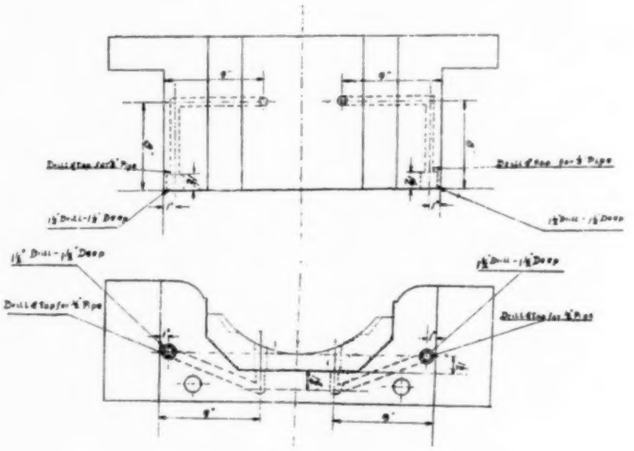


FIG. 17.

We do not wish to offend, but there are times when we think that those who have to do with the designing, building and buying of steel mill equipment, either forget lubrication altogether, or think of it last. Three or four years ago one of the most modern mills of its type ever built was ready to be set up before anyone outside of the general master mechanic apparently gave any serious consideration as to how it was to be lubricated. Among the group was a large drive. The base of the housing was large enough to provide for 15 to 20 barrels of lubricant, far more than required, but the gears were enclosed in a space that would not hold more than a barrel and a half of lubricant, making it necessary to build a tank outside of the housing to carry the necessary volume of lubricant to adequately service that unit.

Again; we were interested in the lubrication of several 2500 H.P. pinions, frequently overloaded 100% or more. Provisions were made for but 35 to 40 gallons of lubricant for these units, which was drawn from the bottom of the housing by a pump

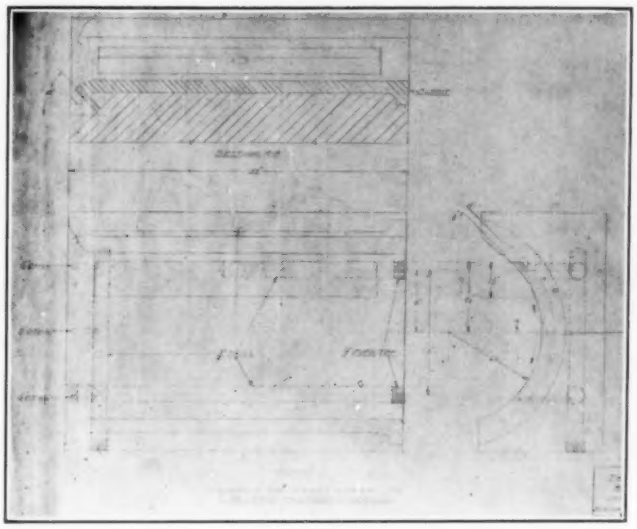


FIG. 18.

and pumped into the gears. A tarry residuum, recommended by the manufacturers of the equipment had to be heated to 200 to 220° F. to pump it. Naturally it wasn't long before the gears became badly cut and scored and pitted. In time the mill provided tanks holding four or five drums of lubricant for each unit which improved the condition a great deal.

We believe there should be closer co-operation between all the engineering and operating organizations having to do with the steel mill industry. By all means the general master mechanics and their organizations, those who are responsible for the maintenance and uninterrupted operation of the mills, should have a voice in the deliberations of the various organizations. I believe, too, that members of the general master mechanic's organization who show themselves worthy, should be permitted to visit other plants and become more familiar with their operating practices.

I earnestly believe that basically some of the difficulties of operation and maintenance originate with the first conception of a new mill. The mill executives decide that they need a new mill. The engineering department is given instructions to work up the layout. Finally the machinery builders are called in to get their ideas and estimates of cost. New schemes are worked out between the mill engineers and the machinery engineer or salesman. Radical departures, probably from anything previous—just a theory oftimes. (It has to be a theory where there is no practice). Finally the mill is built and shipped to the plant to be set up. All this time the men who have to keep the mill running have had little, if anything, to say about how it is going to be lubricated. Oft-times the men erecting the mill and who probably are directly or indirectly connected with the department in charge of oper-

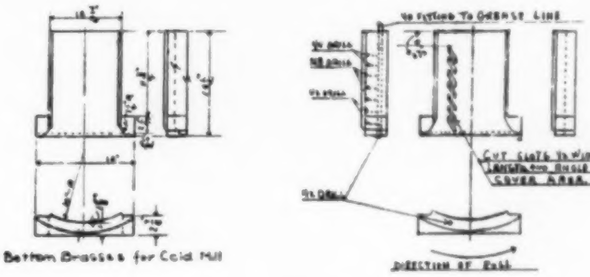
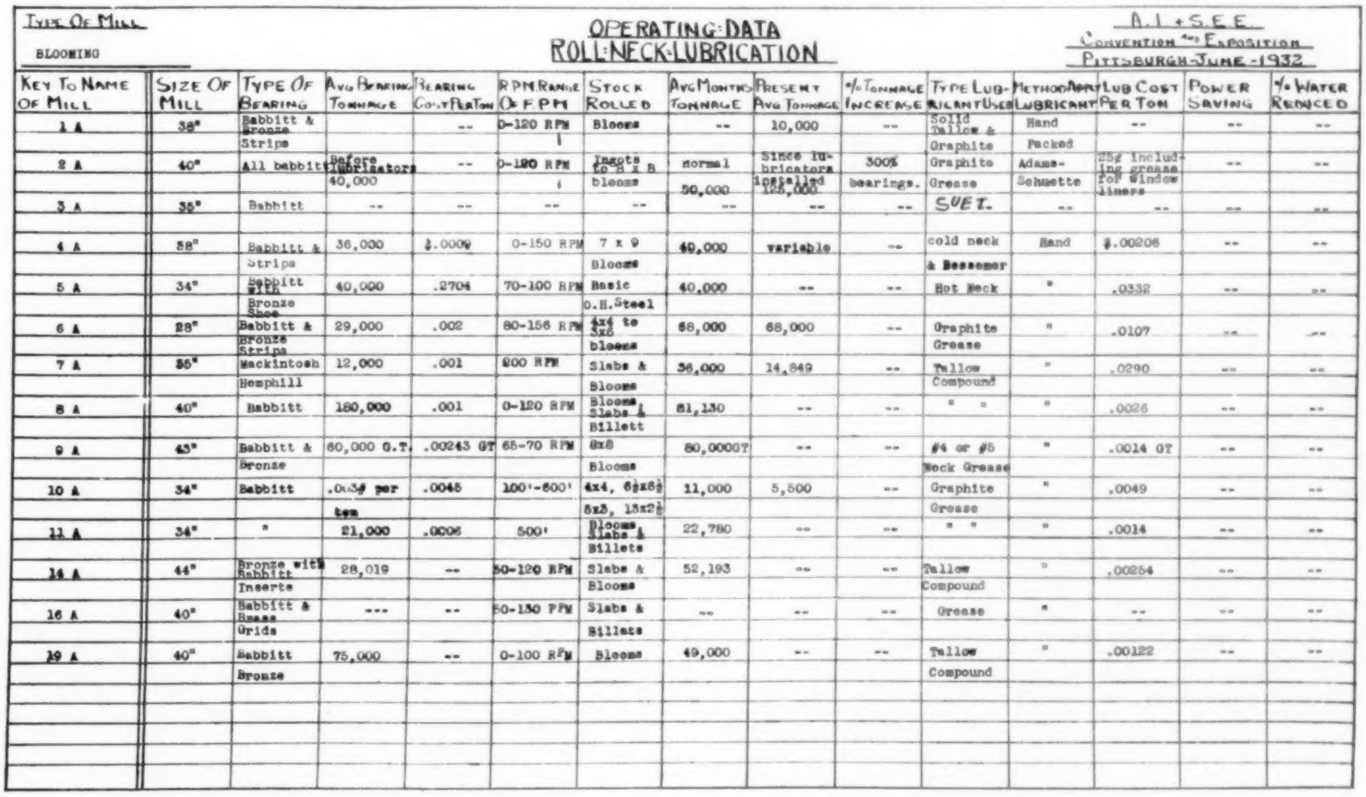


FIG. 19.



Type Of Mill
BLOOMING

OPERATING DATA
ROLL-NECK-LUBRICATION

A.I.&S.E.E.
Convention and Exposition
Pittsburgh-June-1932

Key To Name Of Mill	Size Of Mill	Type Of Bearing	Avg Bearing Tonnage	Bearing Cost Per Ton	RPM Range Or F P M	Stock Rolled	Avg Monthly Tonnage	Present Avg Tonnage	% Tonnage Increase	Type Lubricant Used	Method Applying Lubricant	Lub Cost Per Ton	Power Saving	% Water Reduced
1 A	38"	Babbitt & Bronze		--	0-120 RPM	Blooms	--	10,000	--	Solid Tallow &	Hand	--	--	--
		Strips								Graphite	Packed			
2 A	40"	All babbitt	Before lubricators	--	0-120 RPM	Ingots to 8 x 8	normal	Since lubricators	300%	Graphite	Adams-	25¢ including grease	--	--
			40,000			blooms	50,000	installed 125,000	bearings.	Grease	Schmette	for window liners		
3 A	35"	Babbitt	--	--	--	--	--	--	--	SUET.	--	--	--	--
4 A	58"	Babbitt &	36,000	$.0009	0-150 RPM	7 x 9	40,000	variable	--	cold neck	Hand	$.00206	--	--
		Strips				Blooms				& Bessemer				
5 A	34"	Babbitt with	40,000	.2704	70-100 RPM	Basic	40,000	--	--	Hot Neck	"	.0332	--	--
		Bronze Shoe				O.H. Steel								
6 A	28"	Babbitt &	29,000	.002	80-156 RPM	4x4 to 5x6	68,000	68,000	--	Graphite	"	.0107	--	--
		Bronze Strips				blooms				Grease				
7 A	35"	Mackintosh	12,000	.001	200 RPM	Slabs &	36,000	14,849	--	Tallow	"	.0290	--	--
		Hemphill				Blooms				Compound				
8 A	40"	Babbitt	180,000	.001	0-120 RPM	Blooms, Slabs &	81,130	--	--	" "	"	.0026	--	--
						Billett								
9 A	45"	Babbitt &	60,000 G.T.	.00243 GT	65-70 RPM	8x8	80,000GT	--	--	#4 or #5	"	.0014 GT	--	--
		Bronze				Blooms				Neck Grease				
10 A	34"	Babbitt	.003¢ per	.0045	100'-600'	4x4, 6½x6½	11,000	5,500	--	Graphite	"	.0049	--	--
			ton			8x8, 15x2½				Grease				
11 A	34"	"	21,000	.0006	500'	Blooms, Slabs &	22,780	--	--	" "	"	.0014	--	--
						Billets								
14 A	44"	Bronze with Babbitt	28,019	--	50-120 RPM	Slabs &	52,193	--	--	Tallow	"	.00254	--	--
		Inserts				Blooms				Compound				
16 A	40"	Babbitt & Brass	---	--	50-130 RPM	Slabs &	--	--	--	Grease	"	--	--	--
		Grids				Billets								
19 A	40"	Babbitt	75,000	--	0-100 RPM	Blooms	49,000	--	--	Tallow	"	.00122	--	--
		Bronze								Compound				

FIG. 20.

ation, have to work out changes so they will know it can be operated—lubricated most efficiently.

The machinery builder is required to guarantee the operation of the mill, therefore as the responsibility is his, he elects and is required to say what lubricant should be used. In the meantime, the master mechanic and his organization stand on the side lines hoping that it will not be too long before they can make whatever changes may be necessary, to obtain the best results.

Mr. B. S. Burrell, in his paper previously referred to, makes a complete statement of the gains to be made in maintenance and operation costs as follows:

> "The incentive system is in action with all departmental mechanical and electrical forces participating. Approximately a 10% bonus is paid out to these men for savings in repairs and maintenance costs effected below the amount of money set as an allowable expenditure per ton produced. The Mechanical and Electrical Repair and Maintenance Shops are also operated on an incentive basis.
>
> "Following figures are submitted in reductions relative to lubrication and total Repair and Maintenance costs effected in our Indiana Harbor Plant comparing year 1925 with 1929:
>
> "Reduction in Brass Bearing Costs—25%.
>
> "Reduction in Babbit Bearing Costs—23½%.
>
> "Less money spent for lubrication—40.9%.
>
> "Reduction in Cost per ton for lubrication—52.6%.
>
> "Less money spent for labor and material—13.6%.
>
> "Reduction in cost per ton Labor and Material—30%.
>
> "Increase in tonnage produced—24.2%.
>
> "It pays to use high grade products with the proper degree of economy. The best Repair and Maintenance costs and lowest power consumption per ton of Roll Products shipped is attained only by careful study of each problem and good, sound, common sense and judgment used at all times, not being swayed by good salesmanship of a so called good article in lubricating material, but by sound judgment on the part of your own organization."

There are five groups with whom we should be able to work for an improvement in roll neck lubrication.

1. Operating officials and their mechanical departments.
2. Mill owners.
3. Machinery builders.
4. Manufacturers of bearings and metals.
5. Lubricant manufacturers.

I name them in the order of their importance to the end desired as I see it, and offer some suggestions in connection therewith.

1. The operating officials and their mechanical departments.
   a. They should be kept in closer contact with mill design and operation, especially during the consideration of new equipment. They should be permitted to visit other plants as often as conditions will permit. They should be provided with sources of mechanical data.
   b. When not satisfied that they are obtaining the best results, they should be permitted to obtain supplies, at least for development and experimental periods, from any source they might elect, without interference. If mechanical men are not trustworthy enough to have this authority, they are not trustworthy enough to have in their charge millions of dollars worth of equipment.
2. The Mill Owner.
   a. When new projects are under consideration, or major accessories are to be bought, leave nothing undone to make certain that other conditions warrant their purchase.
   b. Not to expect more from the equipment than it was designed to do, and was agreed upon.
   c. If lubrication is required, make certain that means are provided for the use of the most suitable lubricant in the most efficient manner. Do not let the cost of the more efficient lubricating systems preclude their purchase, or the cost of pattern changes and a few more thousand pounds of castings prevent adequate space being provided for a sufficient volume of lubricant.
   d. Do not let trade relations interfere with the selection of the lubricants to be used. You hold the mechanical department responsible for the operation of the equipment, do not put obstacles in their way.
3. The Machinery Builders.
   a. Provide means for more efficient lubrication of roll necks. Break away from the practices of 200 years ago to a further extent.
   b. Let your designing and mechanical engineers get in closer touch with the men in the mills.
   c. Give more attention to lubrication. Know of your own knowledge what lubricant will be needed for the most efficient operation of the equipment that you manufacture.
4. Bearing Manufacturers.
   a. Determine if the prospective installation is suitable for the application of the type of bearing under consideration.
   b. Be certain that space available will accommodate the size of unit required for the job.
   c. Insist that the loads, speeds and tonnage are not in excess of what the unit was designed for and sold to do.
   d. Give more thought to the proper lubrication of roll neck bearings.
5. The Lubricant Manufacturer.
   a. Let there be a better understanding of the requirements and closer attention to the demands by the maker's technical staff. See that they get out in the mills with their overalls and get their hands into it.
   b. Train sales representatives technically. Give them something besides price books and ability to entertain.

CONCLUSION

Reviewing the foregoing:—

a. It is apparent that the greatest attention is being given to roller bearings. Large sums are being expended for them. There seems to be no comparative data of their operation under similar conditions.

b. Comparatively no improvement has been made in two hundred years of the simplest, least costly types of bearings, the standard open neck and full sleeve babbitt and bronze bearings. Where real efforts have been made to improve them, the results have been outstanding and the cost practically nil. Gains are indicated through a more careful selection of metals on a performance basis rather than price per lb. There is not a plant operating (or ready to do so) that does not have equipment of this type that could be made more efficient.

c. A more thorough understanding as to design and finish of bearings and properly lubricated, seem to be the most direct path to economies in roll neck lubrication.

d. There has been little development and improvement in lubricants by the larger oil interests. "Run of refinery" products and tarry shields, sold largely through trade relations, of the earliest days of refinery practices are used in the greatest volume. Suet is still used.

e. Lubricant manufacturing specialists, some of them in existence for 35 to 50 years, have accomplished the most in the development of more suitable products for the major demands of roll neck lubrication.

f. The grease paddle has been a very expensive tool. The human element is always weak. Both the paddle and the human element should be replaced with mechanical lubrication. There are systems to be had that will apply all kinds of suitable greases in measured amounts, from a few drops to ounces, at a number of predetermined periods, from seconds to hours, that are adapted to all types of roll neck lubrication.

In closing I wish to most heartily thank each and every one who has assisted me in obtaining the data presented here, and for the hearty cooperation of the officials and members of the Association of Iron & Steel Electrical Engineers. I appreciate also the consideration that has been extended to me by you in this presentation.

▲

Compensated Motors and Generators

▲

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and
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To be presented before A. I. & S. E. E. Annual Convention in Pittsburgh, June 20-23, 1932.

The past few years have witnessed an increasing use of direct-current motors and generators in the steel industry. Reversing and continuous mills are producing a considerable variety of products with speeds and refinements that were considered impossible a few years ago. This greater speed of production and variety of refined products have made it desirable to employ electric drives having considerable flexibility. At the same time, close voltage regulation in generators and close speed regulation in motors are needed. Direct-current motors of the adjustable-speed type have often been selected to insure the necessary flexibility and refinements in speed adjustments in connection with modern high-speed mills. The compensated generator and motor have been a very important factor in these developments. This fact, as indicating the general trend, makes it worth while to consider in some detail the characteristics and application of compensated machines.

The direct-current compensated type of machine is provided with a special winding in the pole faces. The method employed is to slot the pole faces and to place a winding about the commutating pole as a center with a part of the winding distributed in the pole faces. The extra winding necessary to produce a commutating flux is concentrated on the commutating poles located midway between the main poles. This winding is connected in series with the armature winding. In the construction of a machine, the compensating winding is placed as near to the armature winding as possible and arranged in such a manner that they carry load current in opposite directions.

The function of a compensating winding is twofold:

First, to create a commutating field which will vary in exact proportion to the armature current. This variation produces the correct condition for sparkless commutation at all loads.

Second, to prevent the magnetic field produced by the armature conductors when carrying current from distorting and cross-demagnetizing on the main shunt field, which action has an important influence upon the performance characteristics of a machine.

Some of the benefits obtained by applying compensating windings to motors and generators can probably be best understood by a study of how the voltage is distributed about a commutator, how the maximum voltage distributed between two adjacent commutator bars is effected by armature reaction on non-compensated machines, and how this voltage is effected by the compensating winding.

The effect of the armature magnetic field (armature reaction) upon the field form of a commutating-pole non-compensated machine is to distort it from a uniform distribution as shown in Fig. 1 (no-load field form), to some shape as shown in Fig. 2 (load field form). The amount of distortion of the load field form varies widely in different designs and depends upon design proportions. The field form distortion will be more pronounced when the armature ampere-turns are high as compared to the shunt field ampere-less turns.

By applying compensating windings to generators or motors, the distortion of the field flux caused by armature reaction is balanced out or neutralized by a flux of equal and opposite strength when the armature ampere-turns and compensating ampere-turns are equal. The more uniformly the compensating winding is distributed about the space surrounding the armature the more perfectly its function is performed.

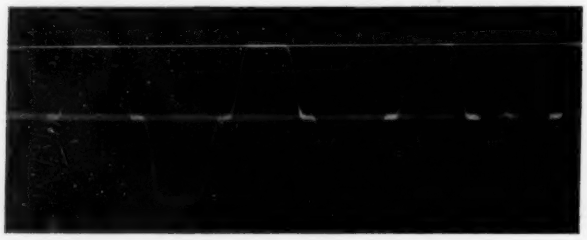


FIG. 1—Oscillograph Record of Field Form, Non-Compensated Generator at No-Load, Indicating the Average Voltage Developed.

The oscillograph records of no-load and full-load field forms shown in Figs. 3 and 4 are from a compensated generator. By a comparison of these oscillograph records it will be observed that the field form distribution is substantially unchanged due to load current, except for the slightest distortion between the compensating winding slots.

In the non-compensated machine, the load field form is distorted (Fig. 2) so that the voltage distributed about the commutator between adjacent commutator bars is not uniform. The height of the full-load field form above the no-load field form is increased due to distortion. The heavier the overload, the greater this distortion is likely to be, resulting in excessive voltage between adjacent bars, which has a bearing on the commutating qualities, and likelihood of flashing. In a compensated machine, the distortion is eliminated, which permits use of higher average voltage between commutator bars. It is to be noted that it is the maximum voltage between bars which is the criterion of flashing between bars and not the average voltage. The maximum volts between commutator bars is dependent upon the ratio of pole arc to pole span, the relative increase in maximum voltage between commutator bars due to distortion of the main field flux, and the number of commutator bars between adjacent brush arms.

FIG. 2—Oscillograph Record of Field Form, Non-Compensated Generator at Full Load, Showing Distorted Field. Note Height of Voltage Peak Above the Average.

With the increasing demand for larger size machines, and increased rating at higher speeds, the use of the compensated machine is being extended in order that the more difficult applications may be met with entire satisfaction. It follows directly that as the kilowatt rating and speed of a machine is increased, the number and resistance of the armature coils decrease, for any given voltage. This results in a reduction of the number of commutator bars, and an increase in the current flowing per bar. Since in a compensated machine the effect of armature reaction on the field is controlled, thereby permitting the use of fewer commutator bars between brush arms, the compensated machine best meets the field of application where higher speeds and higher frequencies are encountered.

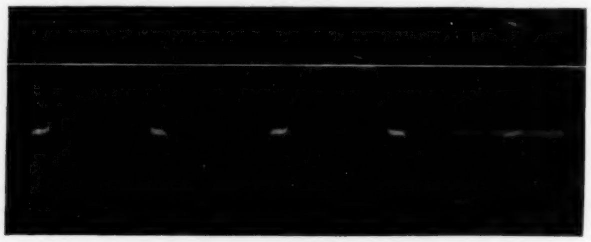


FIG. 3—Oscillograph Record of Field Form, Compensated Generator at No Load.

By taking advantage of these points, it becomes possible by the aid of compensation to make motors and generators which have higher average voltage but no higher maximum voltage per commutator bar, to increase the output capacity, to secure reliable operating performance at higher speeds, closer voltage regulation on generators, closer speed regulation on motors, motors which will operate successfully through wider ranges of speed and commutate successfully high overload currents.

Machines with compensating windings in the pole faces inherently give a straight-line voltage regulation curve. In Fig. 5 is shown a voltage regulation curve from a 1000 KW, 250 volt, 500 RPM compensated generator, and in Fig. 6 is shown a series of voltage curves from a 480 KW, 50 to 150 volt, 750 RPM compensated generator.

There are four general methods of accomplishing speed adjustments on D.C. motors:

FIG. 4—Oscillograph Record of Field Form, Compensated Generator at Full Load. Due to Compensation, the Height and Shape of Field Form Has Not Changed Under Load.

1—Adjusting the motor shunt field strength.
2—Variation of the impressed armature voltage.

3—Inserting and controlling resistance in the armature circuit.

4—A combination of variable voltage and adjustment of the motor field strength.

The first of these methods is simplest and most widely used as it requires only a motor field rheostat in addition to the equipment ordinarily used with a constant speed drive.

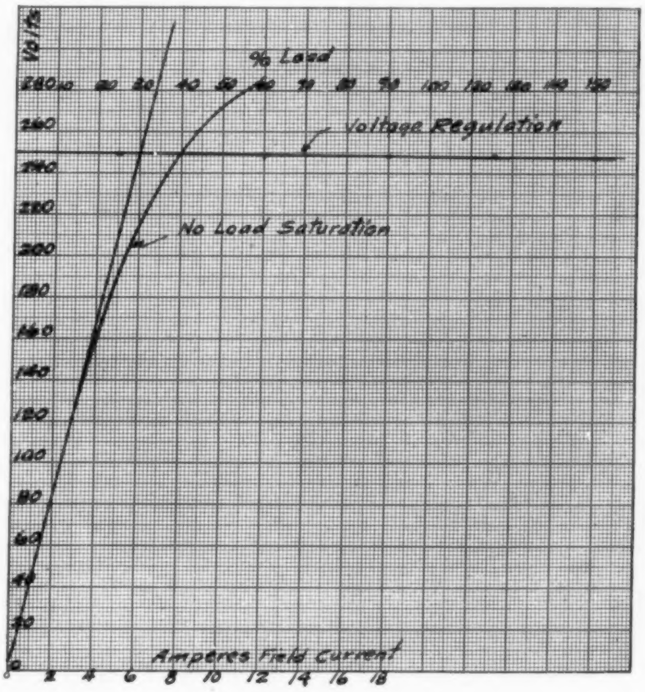


FIG. 5—Voltage Regulation Curve on a 1000-Kw., 250-Volt, 500-R.p.m. Compensated Generator.

For driving reversing mills requiring large powers, D.C. motors are the practical solution of the problem, and the Ward-Leonard system gives much quicker reversal and wastes far less energy than any other system. In this system a separate generator is usually required for each motor. The motor field remains constant and the voltage supplied by the generator is varied by changing the generator field, reducing it from a maximum in one direction to zero, reversing it and building up to a maximum in the other. The ideal reversing cycle for this system involves re-generative braking, which is an inherent part of the system. With the motor running at full speed, the generator field current is reduced gradually, causing the current to the motor to drop down to zero and then reverse, building up to the required current in the reverse direction. The motor is then acting as a generator and converting the kinetic energy of the motor into electrical energy which flows back to the generator. If the reduction of generator field is continued at the proper rate, a steady flow of current from the motor to the generator may be maintained until the motor armature comes to a standstill.

Reversal and gradual building up of the generator field, will cause a continuance of full-load current between motor and generator and an immediate reversal of the direction of rotation of the motor armature as soon as it stops. Re-generative braking, which is really generator action of the motor, starts at the instant the armature current passes through zero and reverses, and continues until the motor armatures come to a standstill. With a properly designed field control for the generator which will give quick reversal of generator field, it is possible to maintain practically a constant flow of current between armatures of motor and generator during reversal of the motor and acceleration of its armature to maximum speed in the reversed direction. If the maximum speed of reversal is desired, the armature current between generator and motor may be increased to as much as 200% load or even to 400% load by a more rapid field change on the generator.

It should be noted that for an operating cycle such as that just described, it is the generator and **not** the motor which gets the severe commutating conditions. The motor at all times operates with full field and should handle its work with ease, but the generator must properly commutate a current of full load, 200% load, or perhaps as much as 500% load, with **practically no field at all.** No type of machine except the compensated generator can be depended on to successfully meet these extremely severe conditions, and in fact it is unwise to try to operate a reversing Ward-Leonard system where extremely heavy currents are handled and quick reversals are desired with any other than a compensated generator.

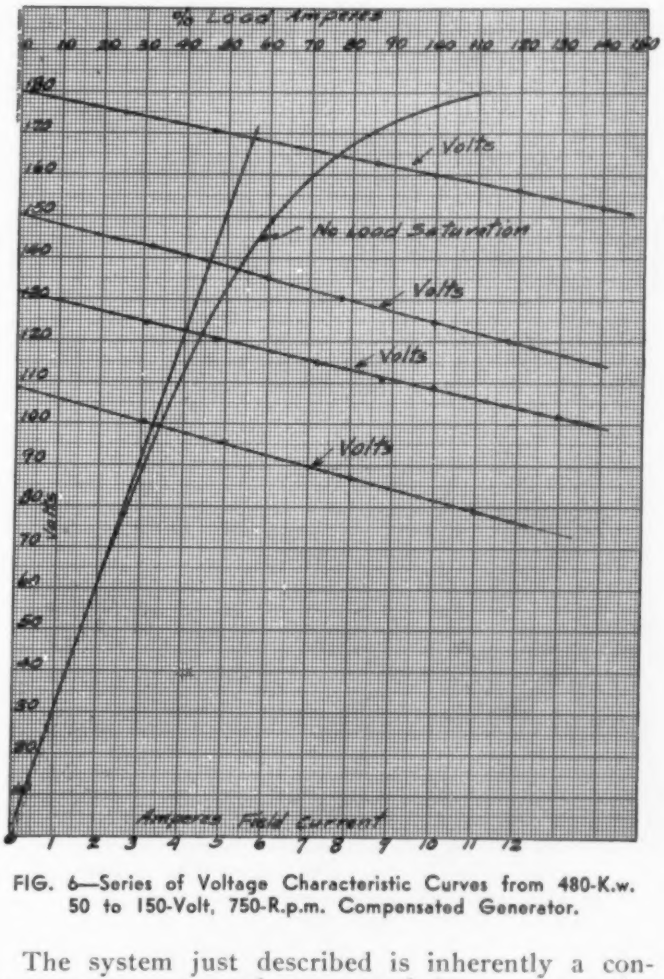


FIG. 6—Series of Voltage Characteristic Curves from 480-K.w. 50 to 150-Volt, 750-R.p.m. Compensated Generator.

The system just described is inherently a constant-torque system, the torque of the motor remaining constant under all speeds as long as the motor armature and field currents remain constant. In mill work it is frequently desirable to speed up the motor beyond its full-field, full-voltage condition. The generator can no longer be depended upon to give this higher motor speed, but by weakening the field of the motor it is easily accomplished. It is quite customary to raise the motor speed as much as 200% in this way. At these increased speeds, the normal capacity of the motor remains at constant horsepower output, inasmuch as the decreasing field and increasing speed balance each other to keep the output constant and the horsepower unchanged.

While it is possible to obtain a 3 to 1 speed range on a non-compensated type of motor, such a machine is likely to commutate very badly under heavy overloads with weak fields. Where loads of this kind are to be carried, and generally where field weakening of the motor is combined with a Ward-Leonard reversing system, it is certainly advisable to use a compensated motor.

The adaptability of the compensated machine is well illustrated by its use for main driving motors on submarines. Here motors of 1000 or 1200 H.P. rating carry frequent overloads of three times their normal rating with fields weakened to less than one-third full magnetic strength, for a period of half an hour at a time. Such a load is enormously more difficult to handle than considerably heavier loads for short periods. This type of motor has been unusually successful in handling these extremely heavy overloads in submarine practice.

As a further explanation, or example, of the ability of the compensated machine to handle severe loads with weak fields,—many years ago a considerable number of storage battery regulating boosters of this type were built and in nearly every case the short peak overload required was enormously high. In one case, the load consisted of unbalanced ore unloaders and the booster furnished for this work was rated at 1800 amperes with a guaranteed peak overload of 9600 amperes for 5 minutes. This machine was tested in the shops of the builders with overloads of 10,000 amperes for periods as long as 10 minutes with successful commutation. The machine had an unusually long life for this class of service and when inspected after 18 years of use the commutator was as perfect as when first built and had every appearance of never having been turned or even sand-papered. It is characteristic of storage battery boosters that at times such a machine must operate as a motor, and it not infrequently occurs that the booster is required to carry loads as high as full-rated load or even more, **with no field whatever.** This is an extreme test of commutation, and nothing but a compensated machine could possibly handle it with entire satisfaction.

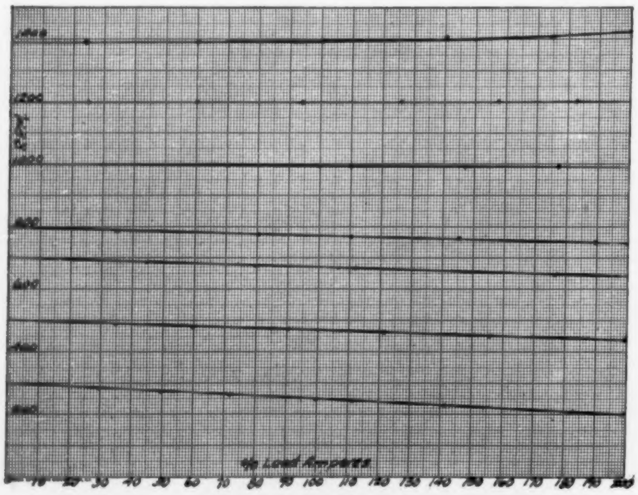


FIG. 7—Speed Characteristic Curves of 50-Hp., 230-Volt, 182-Ampere, 250 to 1400-R.p.m. Compensated Motor.

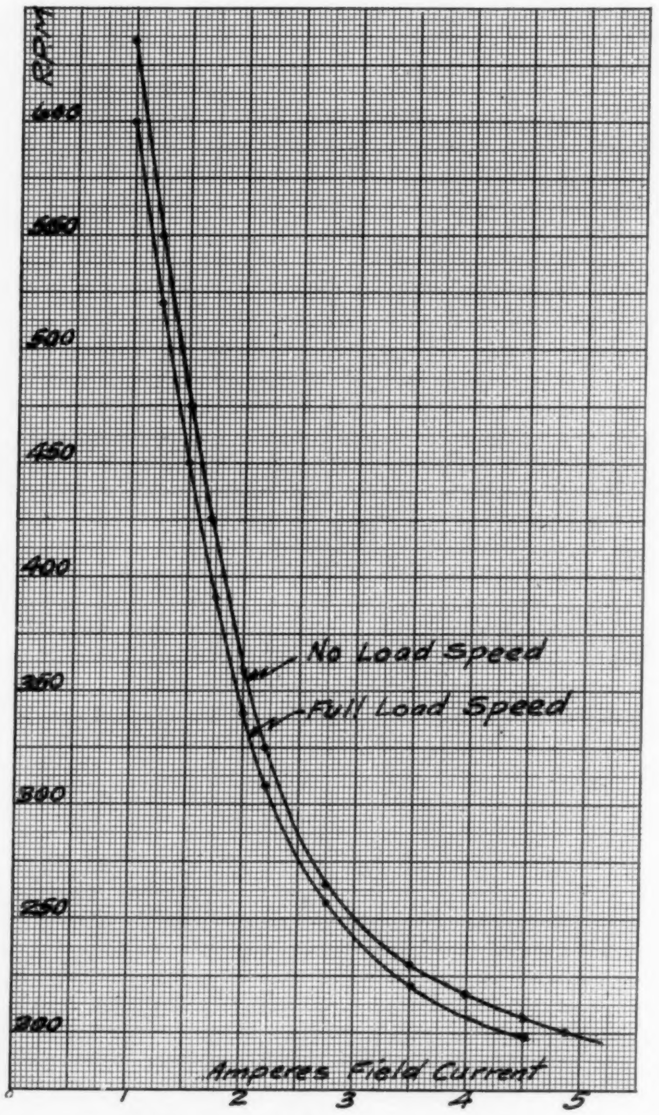


FIG. 8—Speed Characteristic Curves from 1000-Hp., 600-Volt, 200 to 600 R.p.m. Compensated Motor.

In mill work, where adjustable speed is used, it is highly desirable to have motors which not only commutate heavy overloads successfully, but which also, when adjusted for a given speed, will maintain that speed as nearly constant as possible under widely fluctuating loads. The compensated motor fulfills this condition much better than any other type of motor. Fig. 7 shows speed regulation on a 50 H.P. motor built many years ago, and Fig. 8 shows the speed regulation on a rolling mill motor of 1000 H.P., recently constructed. Compensated motors for driving machine tools were built many years ago, having a speed range as high as 10 to 1. These motors were very successful and their commutation, regulation and stability were good. What has already been said in regard to the operation of compensated generators with weakened field and overload applies with equal force to adjustable-speed motors. Closer speed regulation and better stability in a compensated motor follow directly from the prevention of field distortion and may be more clearly understood by a comparison of load characteristics of compensated and non-compensated generators.

For crane service and work of a similar character, the compensated motor is well adapted. Many years ago a shop having 14 cranes was equipped throughout with compensated motors on all cranes. These motors were of the adjustable speed type. Their operation was excellent, but they were very sensitive to movements of the controllers and as a result, crane operators had to be carefully trained to handle this apparatus successfully. This sensitiveness can be removed by the use of a small amount of series winding on the motors which would then be highly effective, particularly for crane hoist motors. This type of motor when used in this way automatically gives re-generative braking while slowing up by strengthening the fields, and dynamic braking by short-circuiting the armature through the starting resistances and finally bringing the armature to a **dead short-circuit.** The first compensated motor ever used on crane work was installed on the 5-ton auxiliary hoist of a 100-ton crane some 28 years ago. This motor and its special controller have been in regular active service from the time it was first installed up to the present time, (the brushes and commutator showing little or no wear). The reason for using this motor was not for experimenting, but because the duty of this hoist required raising or lowering of the load a very small fraction of an inch at a time with precision and certainty. The original series motor installed on this hoist proved far from satisfactory and the compensated adjustable-speed motor met the requirements with perfect ease.

The compensating winding for direct-current machines is by no means a new or untried device. It is forty years since the first machine of this type was built and tested, and since then thousands of compensated machines, from five up to many thousand horsepower capacity, have been put in successful operation. It is not the monopoly of any single builder, since at the present time there are at least five prominent manufacturing companies building the compensated type of machine.

The first machine equipped with compensating windings was designed and built in 1892 by Dr. Harris J. Ryan, at that time Professor of Electrical Engineering at Cornell University and later of Leland-Stanford University. Others may have thought of this same sort of a device before Dr. Ryan, but it has been fairly well established that Dr. Ryan was the first to embody this invention in an actual operative machine and to demonstrate by actual test the value and practicability of the compensating winding. During the next three years several experimental machines with compensating windings were built and tested. In 1894, a 10-KW, 110-volt belted machine was designed and built by the senior writer of this paper. This machine, like all the other compensated machines previously built, used the pole tip for a commutating field. The machine was fitted with metal type brushes. The low resistance of this type of brush made it necessary to shift the brushes with changing load, in order to get sparkless commutation. As a result of the first tests on this 1894 machine, the senior writer of this paper, within a few hours invented the combination of commutating-pole and compensating winding. The compensating winding was carried by a continuous ring, separate from the rest of the field structure. It was, therefore, comparatively easy to build a new ring having commutating poles. This work was completed within the next few weeks, and when re-tested the commutation was perfect with fixed brush position for all loads.

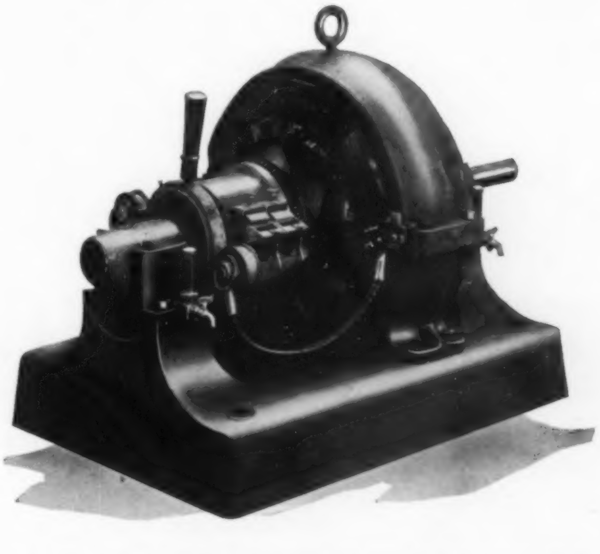


FIG. 9—This Photograph Shows the Original Compensated Generator Built by Mr. M. E. Thompson of Elliott Company's Ridgway Works in 1894. This Machine Was Also the First Generator to Use Commutating Poles and is Therefore a Pioneer in Two Fields.

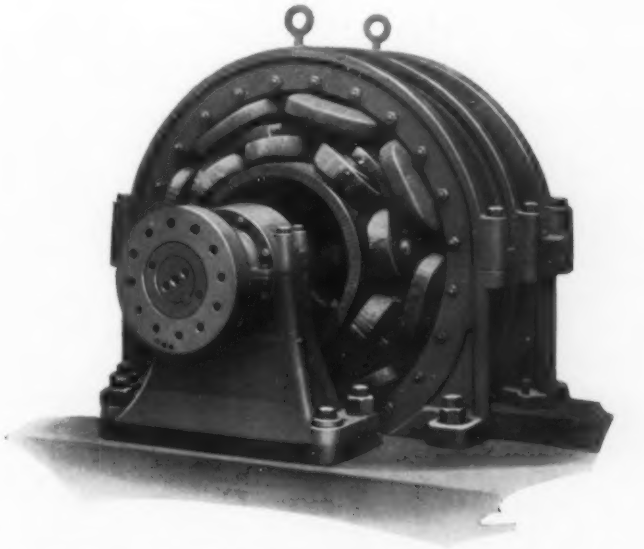


FIG. 10—500 H.p., 600 Volt, 450 to 900 R.p.m. Compensated Direct Current Motor Equipped with Anti-Friction Bearings.

As a result of these later tests, it was immediately evident that the compensated winding to be of practical and commercial value, must perform a two-fold function in creating a fixed and properly proportioned commutating field for all load conditions, and to prevent the distortion of the main field by armature reaction. Within the next year or two this type of machine was regularly on the market in quite a wide variety of speeds and capacities and has been ever since. All commercial machines of the compensated type built in this country or abroad have embodied this double feature of the compensating winding.

It is perhaps worthy of note that within a little more than six months after the successful tests on the 1894 generator, two 200-kilowatt, 550-volt railway generators coupled to Tandem-compound condensing engines, were placed in operation, followed about a year later by two more units of the same design and capacity. These units were in regular daily operation thereafter for a period of more than 25 years.

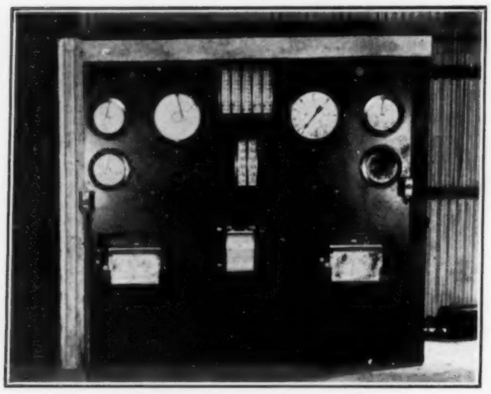
The little 10-KW. experimental generator above mentioned and shown in Fig. 9 is still in existence and has recently been re-conditioned and re-tested for exhibition in Henry Ford's Museum at Dearborn, Mich. None of the original windings have been renewed or changed, although the machine is now 38 years old. This machine was the first compensated commutating-pole generator ever built, and it is probably the first machine of any type on which **commutating poles** were successfully used.

▲

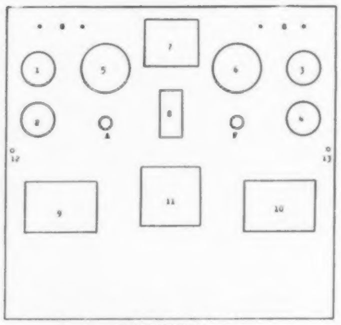
Care and Maintenance of Instruments Used in The Steel Industry

By MARTIN J. CONWAY* and R. C. KENNAN**

To be presented at the A. I. & S. E. E., Combustion Engineering Division, Annual Convention in Pittsburgh, June 20-23, 1932.



Typical Open Hearth Panel Board Installation at Lukens Steel Company.



PANEL BOARD LAYOUT

1-Recording Gauge, steam pressure to furnace.
2-Recording Gauge, oil pressure to furnace.
3-Temperature Gauge, steam for atomization.
4-Temperature Gauge, oil to furnace.
5-Oil Flow Recorder.
6-Oil Flow Indicator.
7-5-point Indicators showing draft in each checker and just before entering waste heat boiler.
8-2-point Indicator, showing suction on both sides of regulator, these two must balance.
9-2-point Recording and Signalling Potentiometer, recording outside checker temperatures and operating lamp banks indicating temperature by light.
10-2-point Recording and Signalling Potentiometer, recording inside checker temperatures.
11-Duplex Pyrometer and CO2 Recorder, recording flue gas temperature and percent CO2 just before entering Waste Heat boiler.
12-Push button for forced draft fan.
13-Cut out switch for oil pump unit.
14-3-color signalling lamp banks.
A-Armature Control, for controlling speed of Pump motor.
F-Field Control, for controlling speed of Pump.
S-Signal lamps.

It is not so long since, that a pressure gauge was the only instrument in the steam plant just as the shovel was the principal tool. Today steam plant performance and efficiency is practically a measure of the number and application of instruments installed. In the steel industry a few years ago the heat treating department and laboratory were the only instrument locations in the plant and these were more or less viewed with awe, by the uninitiated employee and manager alike. Today, the need of measuring instruments is recognized in every department of the plant engaged in the making, shaping, and treating of steel.

It is not sufficient to purchase an instrument and install it in the production department as one would another pipe fitting. The value derived from the same instrument can be increased by locating it in such a manner as to be easily and conveniently used by the operators, mounting in a neat, well arranged manner, properly lighted and protected from dirt, and, definitely determining if the instruments design characteristics of sensitivity, readability, ruggedness, and accuracy are the best obtainable for that particular application. An illustration of such an installation is suggested by the photograph at the beginning of this paper.

During recent years, instruments have been so universally adopted throughout the Steel Industry that the problem of proper installation, care and maintenance has become a matter of considerable importance. No matter how carefully an instrument may be constructed, it cannot give proper service unless it is correctly installed, properly used and given a reasonable amount of competent attention.

This necessitates the creation of a special department since the instrument man should be of a higher specially educated type than the average plant maintenance mechanic.

A full knowledge of the measuring ability of instruments and of their mechanical or electrical limitations enable the instrument department to carry out its investigation with rapidity and efficiency.

CLASSIFICATION:

To study the service problem, the industry may be divided into two classes as follows:

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*Fuel Engineer, Lukens Steel Company, Coatesville, Pa.

**Manager of Service Department, The Brown Instrument Company, Philadelphia, Pa.

1—Large concerns, having many instruments.
2—Small concerns, having few instruments.

LARGE CONCERNS

Selecting a man:

Large concerns having many instruments often employ a man or corps of men to care for their instruments. The type of men generally found most suitable for this class of work are those who have had some fundamental technical education in electrical engineering or allied subjects and who, by their own natural desire or "bent" have acquired some skill in handling tools and equipment necessary to the repairing of instruments. Technical knowledge, while valuable, is not so important as practical experience.

Training of man:

After selecting a man with the foregoing qualifications, he should be sent (by mutual co-operation of the employer and the instrument manufacturer) to the manufacturing plant of the instrument maker for a course of instructions. Some of the more progressive instrument manufacturers have laid out a prescribed course of training for men sent to them for this purpose. A brief outline for a suggested course giving the average time required is outlined below:

Pyrometric Equipment

1—Study of fundamental principles of thermo-electric pyrometers. Millivolt and potentiometer types.
2—Observe construction and testing of thermocouples.
3—Actual practice in assembling and adjusting instruments.
4—Actual practice in assembling and adjusting of galvanometer systems including replacing pointers, pivots, suspensions, springs, jewels, slide wires, etc.
5—Actual practice in calibration and final adjustment.
6—Actual practice assembly and adjusting of control type meters.

Pyrometric course—Total approximate time—

Cadet Engineers	2 weeks
Experienced men	1 week
Foremen	3 days

Flow Meters

1—Study of theory and design of flow meters.
2—Observe mechanical assembly of orifice or other pressure differential equipment.
3—Actual practice in assembling of pressure differential transmitting device (manometer).
4—Actual practice in assembling and adjusting of meters.
5—Final assembly, calibrating and testing of meters and manometers.
6—Discussion of practical problems and use of Flow Meter Engineering Data Book, with Engineering Department.

Flow Meter course—Total approximate time—

Cadet Engineers	1 week
Experienced men	3 days
Foremen	1 day

CO$_2$ Meters, Resistance Thermometers, Control Valve Mechanisms and Miscellaneous Equipment

1—Discussion of application and information desired.
2—Observe functioning of equipment.
3—Actual practice in adjusting equipment.
4—Discussion of flexibility and problem of application with Engineering Department.

Special Instrument course — Total approximate time—

Cadet Engineers	1 week
Experienced men	3 days
Foremen	1 day

The foregoing schedule of time required is only approximate and depends entirely upon the previous experience and capabilities of the individual as well as the time allotted to instruction work and just what is to be accomplished.

As instrument work requires personnel of highly specialized training and the handling of delicate equipment, this service branch should be organized as a separate unit reporting directly to the plant superintendent or the company official in charge of all technical and scientific endeavours. The duties of such an organization will then be to secure the measured facts by instruments of known accuracy and will minimize haphazard individual opinion.

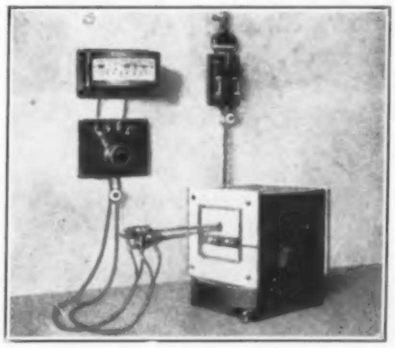

FIG. 1—Checking Thermocouples by the Use of an Electric Muffle Furnace.

After a man has passed through his training course, he should be provided with the necessary tools and equipment, as well as facilities and spare parts to properly handle the work. The following is a suggested layout generally found in a well equipped instrument department.

1—Laboratory Facilities

It is important to bear in mind that a precision instrument may often function improperly due to a slight amount of dust or dirt, presence of iron filings or other seemingly trivial causes. It is, therefore, important that the Laboratory should possess the advantages of good light, free from clouded atmosphere and insofar as possible, free from heavy vibration and shocks frequently encountered in the operation of steel mills. It has been found good practice to see that the room selected is provided with a good ceiling that is relatively dustproof and the interior is painted white to provide the maximum light and to encourage cleanliness.

2—Equipment

(A) The equipment should consist of the following:
1—Suitable work bench.
2—Testing equipment consisting of the following:
  1—portable potentiometer.
  1-battery box with rheostat.
  1—electric muffle type furnace with rheostat.
  1—or more each types of platinum and base metal thermocouples used in the plant. These thermocouples should be certified correct and should be kept on hand as standard for testing only.
  1—set of each type of extension lead wires, portable type for connecting thermocouples to potentiometer.
  1—complete set of temperature millivolt equivalent charts for various types of thermocouples in use.
  1—complete tool kit containing necessary tools.

These can generally be obtained from the instrument manufacturer at a nominal cost. For a permanent installation for checking thermocouples, it is advantageous to have an indicating pyrometer graduated in millivolts mounted with a suitable switch as illustrated in Fig. 1.

(B) The additional equipment required for handling repairs and service on Flow Meters should consist of the following:
1—Test stand with
  1—24″ U tube gauge graduated in inches of water and inches of mercury (which may be detached and used for portable service).
  1—Mercury column 0—30″.
  1—Water column inclined type, range 0—1″ minimum to 0—30″ maximum differential.
  1—Screw pressure pump, capacity 2000 pounds.
  1—Mounting block which should be securely bolted to the floor and provided with necessary mounting bolts to mount manometers for disassembling and cleaning.
  1—Complete set of box type and open-end wrenches.

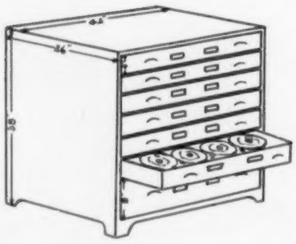

FIG. 2

  1—Set of special tools for making necessary tests and adjustments to meters. These special tools can generally be obtained from instrument manufacturer at a nominal cost.

(C) In addition to the necessary tools and testing equipment, a suitable stock or storage place for parts should be provided. The storage space should generally accommodate spare charts, ink, ribbons, pen, penholders, spare thermocouples, protecting tubes, etc. Ample dustproof filing facilities should be provided to accommodate three months used charts after they have been returned from the production and other departments concerned. A suggestion for such a filing system is shown in figure No. 2.

Duties of Instrument Man

The duties of the instrument man may be briefly defined as follows:
1—Daily inspection of instruments, to remove charts from previous days run and to apply new ones.
2—Properly file charts for future reference.
3—Clean and fill pens of curve drawing ink type instruments, also see that ribbon type recorders are recording properly.
4—Test or interchange thermocouples as frequently as may be found necessary from experience on various processes.
5—Check calibration of instruments at periodic intervals.
6—Make necessary repairs to instruments and thermocouples in laboratory.
7—Supervise installation of all new instrument equipment.
8—Study application of equipment to see that instrument is showing the true condition of the process and is giving the operator the information necessary for the most economical operation of his process. This is an important duty which a properly trained instrument man can handle.

The correct application of the instruments is very important if the benefits sought are to be fully derived.

SMALL CONCERNS

Many of the smaller concerns do not have a sufficient number of instruments in their plant to warrant the employing of an instrument man and must place more dependence upon the service which the instrument manufacturer is able to supply.

Periodic Inspection Service

Some of the leading instrument companies have established a system of periodic inspection service, which is available to users of their equipment at less than standard service rates. Briefly the plan is:
1—A corps of specially trained field service men is maintained at offices in many of the principal cities.
2—The instrument manufacturer enters into an agreement with users of the equipment to have their instruments inspected at periodic intervals of 30, 60, 90 or 180 days.
3—At the agreed time, a field service representative calls at the plant with checking and standardizing equipment. The instruments are checked up and standardized, the wiring and any other portion of the equipment subject to deterioration is examined and if possible repairs are made at that time. If not, the difficulty is reported to the superintendent or man in charge.

4—The trained field service representative is prepared to offer his recommendation for any changes in the application of the equipment, which, based upon his engineering experience, will enable the user to obtain better results.

5—Inspection of equipment at regular periodic intervals will often catch any impending breakdown due to gradual deterioration of the thermocouples or leads which will occur in the course of normal usage. Regular inspection by a trained engineer is highly recommended for plants who do not have an instrument man in their own organization.

EMERGENCY SERVICE FROM FACTORY

Most instrument manufacturers have trained factory service men available for field service upon request. There is necessarily considerable expense in handling emergency service work and it is suggested that a careful study of the problem be made before service is requested from the manufacturer. This study usually results in a saving to the instrument user.

Handling Emergency Service

1—Spare parts.
It is good practice to carry spares to replace parts which are subject to deterioration. The following is a suggested list of items which should be carried on hand for quick replacement.

Pyrometric Equipment

(A) Thermocouple protecting tubes.
(B) Thermocouple elements.
(C) Extension lead wires.
(D) Charts.
(E) Ink
(F) Marking ribbons.
(G) Electric motor for clock drive.
(H) Suspensions for potentiometers.
(I) Galvanometers for potentiometers.
(J) Movable coil for millivoltmeter.
(K) Spare pivots.
(L) Spare jewels.

Thermometers and Pressure Gauges

(A) Sockets or wells for thermometer bulbs
(B) Charts.
(C) Ink.
(D) Clock movement or electric motor if electric clock drive.
(E) Pens and penholders.

Instruction Books

Instrument manufacturers generally supply an instruction book with their equipment. These instruction books have a section devoted to the procedure in checking and adjusting the equipment. The instruction book is of utmost importance in handling emergency service work where the plant does not have an instrument man immediately available. Instruction books are usually available by applying to the manufacturer, being sure to specifically mention the serial number and the model number of the equipment for which such instructions are desired. The logical procedure in handling emergency service is:

1—Consult the Instruction Book.
2—Observe carefully in just what way the equipment does not seem to be performing satisfactorily and attempt to diagnose the difficulty from the instruction book. Very often this will simply be a case of replacement of the thermocouple, cleaning and tightening of the connections, etc., which may be quickly corrected without the delay of writing to the manufacturer and awaiting for his representative to arrive.
3—Do not attempt to make any internal adjustment to the instrument unless you are quite familiar with the entire equipment.
4—If you are unable to locate and remedy the cause of the difficulty, immediately communicate with the manufacturer, giving him the following.
  (A) The serial number of the instrument (check this carefully).
  (B) Explain as fully as you can, just how the instrument performs.

The instrument manufacturer is usually in a position to diagnose the cause of the difficulty and to determine whether the adjustments can be handled by an inexperienced individual or will require a factory service representative.

MODERN INSTRUMENTS

From the viewpoint of the instrument man, interchangeability of parts as well as ruggedness are outstanding features. Within the past few years instrument manufacturers have made some marked advances in instrument design which has greatly increased the flexibility of instruments for modern applications.

1—POTENTIOMETERS

Interchangeability of Parts

Indicating and Recording Potentiometers are now available with readily interchangeable parts so that by substitution of a new slide wire, scale and chart, the instrument can be quickly changed to a different range. This change can be made in an hour or so by an experienced instrument man.

2—RESISTANCE THERMOMETERS

Resistance thermometers of the self-balancing Wheatstone Bridge type are being used more frequently in applications than were formerly handled by distance type of mercury, liquid, vapor tension and gas filled type of thermometers. The advantage of the resistance thermometer type of equipment is that a number of records can be made on one strip chart, also the range can be changed in the field just as easily as the potentiometer type of pyrometer. In case of accident to the lead wires or to the bulb, it is not necessary to return the entire equipment to the manufacturer for repairs. The leads can be repaired right on the job or the bulb can be replaced with a new one with equal facility.

3—MILLIVOLTMETER TYPE

The millivolt type of meter, while not as flexible in the interchangeable range feature, is nevertheless a valuable modern instrument. It lends itself well to general applications where simplicity and ruggedness are important factors. The experienced instrument man will recognize many applications in the steel industry where the millivolt type of meter can be used to advantage.

4—FLOW METERS

Flow meters are offered on the market today which have the interchangeable range feature. This is an item of considerable importance in the steel industry where it frequently becomes necessary to change the location of the meter to meet present day operating conditions. When plants are operating at low capacity, meters will be inefficient under low flow conditions if designed for use under higher normal flow. The meter with the interchangeable range feature can be quickly altered to suit the requirements of the flow.

APPLICATION OF THERMOCOUPLES

Another service which the trained instrument man can render is in the intelligent application of thermocouples. Several of the important considerations are outlined as follows:

1—Importance of Application of Thermocouples

The first consideration is the importance of the particular application in mind. It is false economy to apply a wrought iron protecting tube over the thermocouple if the operating temperature or the atmosphere in which it is installed is such that the tube will require frequent replacement, with the possibility of burning out the thermocouple. On the other hand it would be just as poor economy to apply an expensive thermocouple protecting tube to an application where relatively long life may be obtained with a wrought iron tube. The experienced instrument man will recognize the importance of the application and the relative upkeep expense of thermocouples at that particular point.

2—Life of Thermocouples and Tubes

Within the past several years, users of pyrometric equipment have recognized that the life of thermocouples is largely dependent upon their individual operating conditions. It is almost impossible for the manufacturer to foretell just what life will be obtained from thermocouples. There are so many variable factors such as; operating temperature span from minimum to maximum temperatures, furnace atmosphere, alternate cycles of oxidizing and reducing conditions, etc., which have been brought out in study and development work so that only on very rare occasions and generally by inexperienced individuals, is the manufacturer asked to predict the life of the thermocouple. It is in this study of individual operating conditions that the instrument man is of particular value in the intelligent selection of the proper types of thermocouples and tubes to best handle the job.

3—Special Thermocouple Protecting Tubes

The metal industry has, within the past four or five years, produced a great many metals which may be suitably applied as thermocouple protecting tubes. Each manufacturer claims some special merit for his particular type of metal. There is much to be done in the development of applications where the various types of metal tubes now available can be used. There is a distinct need for classification of the relative merits of tubes for different application. Some research work of this kind has been done and the results tabulated, but the work is still far from being complete. The type of fuel used and sulphur content of the gases of combustion all have a bearing on the ultimate life of the protecting tube.. Pyrometer manufacturers are generally glad to co-operate with users of pyrometric equipment in securing and making up thermocouple protecting tubes and wells from special metals and alloys which are suggested as best for their individual applications. This again is a subject which can be most effectively handled by a good, intelligent instrument man. Generally the user is far better informed of his individual conditions and is in good position to determine whether or not so-called standard protecting tubes are unsuited for his particular requirements. If so, his knowledge of his individual application together with a brief study of the problem should enable him to select some of the special metals that are offered on the open market.

Pyrometer manufacturers usually are in position to provide data and information as to the suitability of certain materials for the purpose. For example, some of the metals are available only in cast form and are not suitable for machining to specifications to withstand high test pressures.

CONCLUSION

It has been well said "Physics is the science of measurement," therefore, measurement is science and the first step to economy. Hence the instrument department is a scientific department and the problem of selection, installation, care and maintenance of measuring instruments should be entrusted to specially trained men who are progressively interested in their work. The problem of care and maintenance, together with the application of instruments in the steel industry is more important, under present highly competitive business conditions than ever before, and by maintenance, we mean good maintenance, for an instrument in error is worse than useless, once the habit of placing reliance on its indications has been established. The company having the proper equipment to measure their entire process, backed up with an intelligent recognition of the necessary care and maintenance of their instruments, will find a substantial return in proportion to their investment, and the amount of time and interest devoted to the subject.

A. I. & S. E. E. Proposed Safety Regulations Necessary When Working on High Tension Lines

A. No work must be done on lines or parts carrying high voltage (440 AC and over) until all proper precautions have been taken for protection of the workmen. Whatever the necessity for work on a high tension line and regardless of who asks for the outage the order shall be sent through some definitely designated person for the plant (preferably the department superintendent or his assistant) who shall have authority to take the line out of service and who shall be familiar with all standard regulation and special conditions for the line and the plant in which it is installed.

B. The plant shall have a complete map of each high tension circuit showing poles, number of lines and special features so that accurate checking will be possible. These maps must be kept up-to-date by requiring the line foreman to report all addition or changes made by him or any of his men, to the department superintendent, who will see that the entrees are made on all the maps that are in service.

C. When a line is to be taken out of service notice shall be given to the station operator in writing. The clearing of the line shall be supervised by the line foreman and checked by the general foreman or department superintendent, a regular routine being followed.

1. All oil switches shall be opened at both ends of the feeder, under the supervision of the line foreman, beginning at the far end of the circuit, special attention being given to loop circuits which normally receive current at two or more points.

2. Following this all disconnects in the circuit shall be pulled, truck type panels pulled and locked out. (Standard locks should be removed by the operator and the line foreman's own locks substituted).

3. Then each wire of the circuit should be tested with some approved type of instrument to determine that it is dead, the testing instrument being checked, before and after each test, against a known live line to insure against false indications due to open circuits in the instrument.

4. The line being shown to be dead should then be carded with danger signs and also tags carrying the name of each man who is to work on the line and signed by the line foreman.

NOTE: The line must not be energized until all men, whose names are entered on this card, are accounted for and authority to return the line to service is given by the line foreman in writing. No men not entered on this card should be allowed on the line.

5. The proper parties at the direction of the line foreman will now ground and short the dead ends of the line.

6. Where more than one gang is to work on the same circuit all safe guarding conditions must be separately complied with by each gang.

D. The line foreman will now check the line against his map from the station and other sources of current to the point where the work is to be done to see that no mistakes have been made and shall have lines tested at the point of work for voltage with some approved instrument, as referred to in (C-3) after which the line shall be grounded and short circuited on each side of the point where the work is to be done. The shortening of the line may be done by any approved method, the commonest being to throw a dry hand line over the wires and then pull a chain, one end having been well grounded, over the wires.

NOTE: The objection to this is that chains have been known to permit a dangerous voltage on the line due to the link to link resistance. This defect may be largely remedied by interweaving flexible copper cable through the chain links. The safest plan is to use some form of positive contact clamp for attaching ground wires, such as can now be bought in the market.

E. After the work on the line is completed the line foreman shall see that all local protective grounds and shorts are removed and that all workmen are off the lines and accounted for and so reported to the station operator, who shall supervise the removing of the danger signs and tags and authorize the station operator, in writing, to reenergize the line.

F. The station operator will remove his grounds and shorts and test the line to see that it is clear before closing his disconnects and oil switches.

G. Preferably all high tension wires should be dead on a pole to be worked on by linemen, but in some main lines one circuit must be left alive. Under these conditions the construction of this line must be such as to show a safe distance between the live line and the one to be worked on and the lineman specially instructed as to the conditions and the danger of approaching the live line.

H. Should the outage carry over from one turn to another the station operator shall advise his relief as to all details calling attention to the names of the men on the line, entering the time of this transfer of responsibility on the danger card over his signature. The relief shall also note the taking over of the responsibility on the danger card.

I. Any change in the personnel of the gangs working on the line must be at once reported to the station operator. If a man leaves the job he must report personally to the station operator and have his name marked off with the time noted, and if a new man is to go on the job he must first report to the station operator and have his name added to the danger tag with the time of such entry.

Report of A. I. & S. E. E. Ladle Crane Committee

By F. W. CRAMER, Chairman

To be presented at A. I. & S. E. E. Annual Convention in Pittsburgh, June 20-23, 1932.

The length of time the main hoist cables can be kept in service on ladle cranes is important not only from a safety standpoint, but also from the economic consideration. These particular cables are probably the most expensive ones in a steel mill to replace and also the ones the maintenance department takes no chances of using when they become worn. It has been very interesting during the past year to investigate the length of life secured on these installations at the various plants and to tabulate the factors entering into their use.

Many years ago when steel cables replaced the chains on crane hoists, the improvement not only in life but also in costs was so great that the applications and care of proper steel cables did not present a serious problem. For some time there has been a prevalent opinion that if a life of one year could be secured on ladle crane cables, it was all that could be expected. This opinion is verified by the answers received from the many steel plants that had a part in this study. The degree of interest in the subject may be measured by the fact that many plants were asked for this information and each and every one returned their data within two weeks. We believe this is a record response in handling matters by a questionnaire.

In comparing the data submitted there are many things that could be pointed out as contributing factors to a short life of these cables.

You will note from the charts that a majority of the cranes are overloaded. It may be that the overloading is even worse than shown, as there is a tendency to figure open hearth hot metal tonnage as short tons, whereas it is common practice to use 2240 lbs. to the ton in these operations. Again, if these figures were not obtained from actual scale tonnages it might be that spouts had been bricked up or the lining changed so that ladles held more than they were originally intended to. Some of the larger cranes that have been purchased during the past five years are shown to be overloaded as much as 40 per cent.

Another interesting fact is that on the later cranes, more thought has been given to the design of sheave wheels and drums to secure the proper diameters. There has also been more thought given to the lay of the cable and the reeving to prevent the ropes from excessive fleet angles or rubbing on each other as they pass. Although the question was not included in the list, we find that the actual construction of drums and sheave wheels has been improved. Sheave wheels with anti-friction bearings are now standard on ladle cranes. The sheave wheels themselves are no longer iron or steel castings with their attendant blow holes whose sharp edges had a tendency to cut and wear the strands of the cables. Factors such as these are contributing to longer cable life.

You will also note that the so-called safety cables are being discontinued and the reeving so designed that all cables are used for lifting. The safety feature is taken care of by the equalizer beam which is designed to equalize the load in case a cable breaks. This also replaces the equalizer sheaves which have been a source of trouble on many ladle cranes. Due to their slight rocking motion they were very difficult to lubricate and were usually located in an inaccessible place.

It was also interesting to find out whether steel center or hemp center cables were being used. Of the 49 plants reporting, 26 use hemp center cables and 23 the steel or soft iron center. From this it is evident that both types of cable are giving satisfaction. The tabulation shows that only in five plants do the cranes hold the ladle while the heats are being tapped. In three of these cases the steel center cables are used. It would be interesting to compare these figures with similar figures taken ten years ago when practically all ladle cranes held the ladles during tapping. The practice should have a bearing on crane life, as the intense heat and the flame both have their effect on the cables.

There does not seem to be any uniform way of applying dressing to the cables and black oil seems to be the favorite dressing to use. The proper dressing and its application is a subject worthy of a separate discussion, and should be presented at one of our meetings. From a few actual observations there are very few ladle cranes that have the dressing properly applied.

There are many other factors entering into cable life that could not be brought out in this study, as they represent individual cases. The graphite condition about mixer and pig caster cranes is more severe in some plants than in others. In one location the direction of the prevailing winds is such as to cause an excessive amount of graphite to be deposited which tends to shorten the cable life. The number of lifts per crane per day is another factor that cannot be determined. This factor would show the ratio of wear from service of the different cranes. Another factor is the proper equalization of loads on each of the cables. The stretch on the individual cables is not always the same, and if the equalizer sheaves are not working properly, or if the adjustments on the load beams are not carefully watched, one cable will receive most of the wear.

Perhaps the most important fact brought out in this study is the fact that in 17 of these plants the normal life of ladle crane cables is less than six months. It may be that in some of these plants this time has been set as a limit, regardless of the condition of the cable. In others it appears as though a study of the crane design is worth while to see what could be done to lengthen the life. The proper cable dressing and its applications should assist in other cases. We feel that modern improvements and practices point to a life of better than a year in this service.

In conclusion, the question, when should the cables be changed, is the one we are trying to solve. The steel industry takes no chances on these cables in any of the plants. The cable manufacturers will offer no set rule for changing, as outward appearances are often misleading, and a cable may be ruined internally and appear to be good on the outside. Fatigue and crystallization are not usually visible, so that judgment, broken strands and wear are the three items the mill man has to base his opinion on when a change should be made.

These few observations are set forth to call to the attention of the industry the practice of our many steel plants in changing and inspecting ladle crane cables. It is hoped that the discussion on this subject will bring out additional information that will assist in obtaining data on the proper time to change cables on ladle cranes.

	PLANT NO. 1	PLANT NO. 2	PLANT NO. 3	PLANT NO. 4	PLANT NO. 5
QUESTION No. 1 LOCATION OF CRANE	ELECTRIC FURNACE	PIG CASTING MACH. BLAST FURNACE	CHARGING FLOOR OPEN HEARTH	CHARGING FLOOR OPEN HEARTH	CHARGING FLOOR OPEN HEARTH
QUESTION No. 2 CAPACITY OF CRANE	30 TONS	35 TONS	40 TONS	40 TONS	40 TONS
QUESTION No. 3 MAXIMUM LIFT OF CRANE IN ACTIVE OPERATION					
(A) WEIGHT OF HOT METAL		60,000-LBS.	60,000-LBS.	55,000-LBS.	55,000-LBS.
(B) WEIGHT OF HOOKS		2,400-LBS.	2,400-LBS.	3,900-LBS.	3,900-LBS.
(C) WEIGHT OF SPREADER BEAM		3,480-LBS.	3,480-LBS.	3,600-LBS.	3,600-LBS.
(D) WEIGHT OF CABLES		300-LBS.	420-LBS.	301-LBS.	243-LBS.
(E) WEIGHT OF LADLE & LINING		34,000-LBS.	34,000-LBS.	27,000-LBS.	27,000-LBS.
(F) TOTAL WEIGHT	64,000-LBS.	100,180-LBS.	100,300-LBS.	89,801-LBS.	89,743-LBS.
QUESTION No. 4 SIZE AND NUMBER OF CABLES	1-CABLE 3/4" DIAMETER	1-CABLE 7/8" DIAMETER	1-CABLE 7/8" DIAMETER	2-CABLES 7/8" DIAMETER 6X19 STRANDING	2-CABLES 3/4" DIAMETER 6X19 STRANDING
QUESTION No. 5 LENGTH OF CABLES		240'-0"	350'-0"	245'-0" EACH	270'-0" EACH
QUESTION No. 6 ARE SAFETY CABLES USED		NO	NO	NO	NO
QUESTION No. 7 DO CABLES HAVE HEMP OR IRON CORD	HEMP	HEMP	HEMP	IRON	HEMP
QUESTION No. 8 DIAMETER OF MAIN SHEAVES	16"	26 1/4"	26 1/4"	26 1/4"	26 1/4"
QUESTION No. 9 DIAMETER OF EQUALIZER SHEAVES	11"	26 1/4"	26 1/4"	NONE	NONE
QUESTION No. 10 DIAMETER OF MAIN HOIST DRUM	18 5/8"	26 1/4"	26 1/4"	29 3/8"	26 1/4"
QUESTION No. 11 ARE HOOKS SOLID OR LAMINATED	SOLID	LAMINATED	LAMINATED	LAMINATED	LAMINATED
QUESTION No. 12 INSPECTION OF CABLES					
(A) WHEN INSPECTED				EVERY DAY	EVERY DAY
(B) BY WHOM INSPECTED	BY CRANE INSPECTOR			FOREMAN OF THIS DEPARTMENT	FOREMAN OF THIS DEPARTMENT
(C) HOW INSPECTED	CLEANS & INSPECTS SECTION OF CABLE			WITH HEAVY GLOVE AND PIECE OF WASTE	WITH HEAVY GLOVE AND PIECE OF WASTE
QUESTION No. 13 IS TEST LIFT MADE TO ASCERTAIN IF BRAKES ARE HOLDING BEFORE TAKING HEAT AWAY FROM STAND	NO	YES	YES	YES	YES
QUESTION No. 14 AVERAGE LIFE OF CABLES ON THIS CRANE	6 MONTHS	3 MONTHS	4 MONTHS	5 1/2 MONTHS	11 MONTHS
QUESTION No. 15 MAXIMUM LENGTH OF TIME CABLES ARE LEFT IN SERVICE ON THIS CRANE		5 MONTHS	6 MONTHS	8 MONTHS	12 MONTHS
QUESTION No. 16 TYPE OF LUBRICANT USED ON CABLES AND HOW IS IT APPLIED	BLACK OIL POURED ON DRUM			BLACK OIL APPLIED WITH A PAINT BRUSH	BLACK OIL APPLIED WITH A PAINT BRUSH
QUESTION No. 17 DOES CRANE HAVE TO HOLD LADLE WHILE HEAT IS BEING TAPPED OR WHILE ADDITIONS ARE BEING MADE IN LADLE	YES	NO	NO	NO	NO
QUESTION No. 18 REMARKS				AUX. HOIST 1-CABLE 5/8" DIAMETER 200'-0" LONG	AUX. HOIST 1-CABLE 5/8" DIAMETER 200'-0" LONG

	PLANT NO. 6	PLANT NO. 7	PLANT NO. 8	PLANT NO. 9	PLANT NO. 10
QUESTION No. 1 LOCATION OF CRANE	ELECTRIC FURNACE DEPARTMENT	PIG CASTING MACH FURNACE DIVISION	CHARGING FLOOR OPEN HEARTH	METAL MIXER OPEN HEARTH	METAL MIXER BESSEMER
QUESTION No. 2 CAPACITY OF CRANE	50 TONS	60 TONS	75 TONS	75 TONS	75 TONS
QUESTION No. 3 MAXIMUM LIFT OF CRANE IN ACTIVE OPERATION					
(A) WEIGHT OF HOT METAL			84,000-LBS.	84,000-LBS.	90,000-LBS.
(B) WEIGHT OF HOOKS			8,000-LBS.	8,000-LBS.	7,000-LBS.
(C) WEIGHT OF SPREADER BEAM			12,000-LBS.	12,000-LBS.	12,000-LBS.
(D) WEIGHT OF CABLES			1,000-LBS.	2,000-LBS.	3,300-LBS.
(E) WEIGHT OF LADLE & LINING			30,000-LBS.	30,000-LBS.	32,000-LBS.
(F) TOTAL WEIGHT	64,000-LBS.	120,000-LBS.	135,000-LBS.	136,000-LBS.	144,300-LBS.
QUESTION No. 4 SIZE AND NUMBER OF CABLES	1-CABLE 1" DIAMETER	1-CABLE 1" DIAMETER 6X19 STRANDING PLOW STEEL	2-CABLES 1" DIAMETER 6X37 STRANDING	2-CABLES 1" DIAMETER 6X37 STRANDING	6-CABLES 1" DIAMETER
QUESTION No. 5 LENGTH OF CABLES			280'-0" EACH	430'-0" EACH	2 LENGTHS-530'-0" EACH 4 LENGTHS-265'-0" EACH
QUESTION No. 6 ARE SAFETY CABLES USED			NO	NO	NO
QUESTION No. 7 DO CABLES HAVE HEMP OR IRON CORD	HEMP	IRON	HEMP	HEMP	IRON
QUESTION No. 8 DIAMETER OF MAIN SHEAVES	32"	36"	37"	37"	30" 32"
QUESTION No. 9 DIAMETER OF EQUALIZER SHEAVES	10 3/4"	12"	NONE	NONE	15"
QUESTION No. 10 DIAMETER OF MAIN HOIST DRUM	30"	30"	38"	38"	53"
QUESTION No. 11 ARE HOOKS SOLID OR LAMINATED	SOLID	SOLID	LAMINATED	LAMINATED	LAMINATED
QUESTION No. 12 INSPECTION OF CABLES (A) WHEN INSPECTED (B) BY WHOM INSPECTED (C) HOW INSPECTED	BY CRANE INSPECTOR CLEANS AND INSPECTS SECTION OF CABLE	ONCE A MONTH MECHANICAL DEPARTMENT	DAILY BY MOTOR INSPECTOR AND WEEKLY BY TURN FOREMAN	DAILY BY MOTOR INSPECTOR AND WEEKLY BY TURN FOREMAN	DAILY BY TURN FOREMAN FROM CRANE CAB OR TOP FLOOR IN BUILDING
QUESTION No. 13 IS TEST LIFT MADE TO ASCERTAIN IF BRAKES ARE HOLDING BEFORE TAKING HEAT AWAY FROM STAND	NO	NO	YES	YES	YES
QUESTION No. 14 AVERAGE LIFE OF CABLES ON THIS CRANE	6 MONTHS	6 MONTHS	10 MONTHS	10 MONTHS	9 MONTHS
QUESTION No. 15 MAXIMUM LENGTH OF TIME CABLES ARE LEFT IN SERVICE ON THIS CRANE		18 MONTHS	12 MONTHS	12 MONTHS	12 MONTHS
QUESTION No. 16 TYPE OF LUBRICANT USED ON CABLES AND HOW IS IT APPLIED	BLACK OIL ON DRUM	INSIDE CABLE SHIELD COMPOUND	STANDARD OIL CO. CABLE DRESSING APPLIED BY POURING ON WHILE CABLE RUNS OVER SHEAVES	STANDARD OIL CO. CABLE DRESSING APPLIED BY POURING ON WHILE CABLE RUNS OVER SHEAVES	STANDARD OIL CO. PRODUCT APPLIED WITH A BRUSH
QUESTION No. 17 DOES CRANE HAVE TO HOLD LADLE WHILE HEAT IS BEING TAPPED OR WHILE ADDITIONS ARE BEING MADE IN LADLE	YES	NO	NO	NO	NO
QUESTION No. 18 REMARKS					

	PLANT NO. 11	PLANT NO. 12	PLANT NO. 13	PLANT NO. 14	PLANT NO. 15
QUESTION No. 1 LOCATION OF CRANE	CHARGING FLOOR OPEN HEARTH	CHARGING FLOOR OPEN HEARTH	CHARGING FLOOR OPEN HEARTH	POURING SIDE OPEN HEARTH	POURING SIDE OPEN HEARTH
QUESTION No. 2 CAPACITY OF CRANE	75 TONS	75 TONS	75 TONS	100 TONS	100 TONS
QUESTION No. 3 MAXIMUM LIFT OF CRANE IN ACTIVE OPERATION					
(A) WEIGHT OF HOT METAL	139,000-LBS.	86,000-LBS.		210,000-LBS.	210,000-LBS.
(B) WEIGHT OF HOOKS	14,000-LBS.	6,000-LBS.		5,000-LBS.	5,000-LBS.
(C) WEIGHT OF SPREADER BEAM	14,000-LBS.	10,000-LBS.		10,000-LBS.	5,000-LBS.
(D) WEIGHT OF CABLES	14,000-LBS.	1,000-LBS.		1,000-LBS.	1,000-LBS.
(E) WEIGHT OF LADLE & LINING	56,000-LBS.	44,000-LBS.		86,000-LBS.	86,000-LBS.
(F) TOTAL WEIGHT	209,000-LBS.	147,000-LBS.	150,000-LBS.	312,000-LBS.	307,000-LBS.
QUESTION No. 4 SIZE AND NUMBER OF CABLES	2-CABLES 1" DIAMETER	2-CABLES 7/8" DIAMETER IMPROVED PLOW STEEL 6X37 STRANDING	2-CABLES 1" DIAMETER	2-CABLES 1 1/8" DIAMETER IMPROVED PLOW STEEL 6X37 STRANDING	2-CABLES 7/8" DIAMETER IMPROVED PLOW STEEL 6X37 STRANDING
QUESTION No. 5 LENGTH OF CABLES	450'-0" EACH	445'-0" EACH	275'-0" EACH	272'-6" EACH	420'-0" EACH
QUESTION No. 6 ARE SAFETY CABLES USED	NO	NO		NO	NO
QUESTION No. 7 DO CABLES HAVE HEMP OR IRON CORD	HEMP	HEMP	STEEL CENTER	HEMP	HEMP
QUESTION No. 8 DIAMETER OF MAIN SHEAVES	30"	26 1/4" 28 1/4"	26 1/4"	30"	26 1/4"
QUESTION No. 9 DIAMETER OF EQUALIZER SHEAVES	16"	12"	26 1/4"	30"	26 1/4"
QUESTION No. 10 DIAMETER OF MAIN HOIST DRUM	51"	26 1/4"	34 1/8"	38"	30"
QUESTION No. 11 ARE HOOKS SOLID OR LAMINATED	LAMINATED	LAMINATED	LAMINATED	LAMINATED	LAMINATED
QUESTION No. 12 INSPECTION OF CABLES (A) WHEN INSPECTED (B) BY WHOM INSPECTED (C) HOW INSPECTED	EACH DAY LADLE CRANE INSPECTOR BY RIDING SPREADER	GENERAL INSPECTION DAILY & WEEKLY INSPECTION THOROUGHLY BY O.H. CRANE MECH. RIDING SPREADER HOOK TO EXAMINE CABLES	BY CRANE INSPECTOR	GENERAL INSPECTION DAILY & WEEKLY INSPECTION THOROUGHLY BY O.H. CRANE MECH. RIDING SPREADER HOOK TO EXAMINE CABLES	GENERAL INSPECTION DAILY & WEEKLY INSPECTION THOROUGHLY BY O.H. CRANE MECH. RIDING SPREADER HOOK TO EXAMINE CABLES
QUESTION No. 13 IS TEST LIFT MADE TO ASCERTAIN IF BRAKES ARE HOLDING BEFORE TAKING HEAT AWAY FROM STAND	YES	YES	YES	YES	YES
QUESTION No. 14 AVERAGE LIFE OF CABLES ON THIS CRANE	8 MONTHS	6 MONTHS REGARDLESS OF SHAPE	6 MONTHS	6 MONTHS REGARDLESS OF SHAPE	6 MONTHS REGARDLESS OF SHAPE
QUESTION No. 15 MAXIMUM LENGTH OF TIME CABLES ARE LEFT IN SERVICE ON THIS CRANE	12 MONTHS	6 MONTHS	12 MONTHS	6 MONTHS	6 MONTHS
QUESTION No. 16 TYPE OF LUBRICANT USED ON CABLES AND HOW IS IT APPLIED	STANDARD OIL CABLE GREASE POURED INTO DRUM GROOVES AS CABLE IS COILED ON	IRONSIDE CABLE COMPOUND APPLIED HOT TO CABLE ON DRUMS	SPECIAL CABLE COMPOUND	IRONSIDE CABLE COMPOUND APPLIED HOT TO CABLE ON DRUMS	IRONSIDE CABLE COMPOUND APPLIED HOT TO CABLE ON DRUMS
QUESTION No. 17 DOES CRANE HAVE TO HOLD LADLE WHILE HEAT IS BEING TAPPED OR WHILE ADDITIONS ARE BEING MADE IN LADLE	NO	NO	NO	NO	NO
QUESTION No. 18 REMARKS		INSPECTED MONTHLY BY COMMITTEE INCLUDING ELECT., MECH. & SAFETY DEPT. MEN-WRITTEN REPORT MADE AND FILED.		INSPECTED MONTHLY BY COMMITTEE INCLUDING ELECT., MECH. & SAFETY DEPT. MEN-WRITTEN REPORT MADE AND FILED	INSPECTED MONTHLY BY COMMITTEE INCLUDING ELECT., MECH. & SAFETY DEPT. MEN-WRITTEN REPORT MADE AND FILED

	PLANT NO.16	PLANT NO.17	PLANT NO.18	PLANT NO.19	PLANT NO.20
QUESTION No.1 LOCATION OF CRANE	CHARGING BAY OPEN HEARTH	OPEN HEARTH	METAL MIXER BESSEMER	OPEN HEARTH	
QUESTION No.2 CAPACITY OF CRANE	100 TONS	100 TONS	115 TONS MAIN HOIST 75 TONS AUX. HOIST	125 TONS	125 TONS
QUESTION No.3 MAXIMUM LIFT OF CRANE IN ACTIVE OPERATION					
(A) WEIGHT OF HOT METAL	126,000-LBS.	134,400-LBS.	90,000-LBS.	200,000-LBS.	230,000-LBS.
(B) WEIGHT OF HOOKS	8,400-LBS.	6,070-LBS.	8,000-LBS.	8,000-LBS.	8,200-LBS.
(C) WEIGHT OF SPREADER BEAM	22,300-LBS.	6,000-LBS.	6,000-LBS.		24,200-LBS.
(D) WEIGHT OF CABLES	2,750-LBS.	849-LBS.	3,000-LBS.		1,400-LBS.
(E) WEIGHT OF LADLE & LINING	59,000-LBS.	47,000-LBS.	40,000-LBS.	72,000-LBS.	65,000-LBS.
(F) TOTAL WEIGHT	218,450-LBS.	194,319-LBS.	147,000-LBS.		328,800-LBS.
QUESTION No.4 SIZE AND NUMBER OF CABLES	4-CABLES 1" DIAMETER REEVED IN 28 PARTS	22-CABLES 1⅛" DIAMETER	MAIN CABLE 1⅛" DIAMETER AUX. CABLE 1" DIAMETER	16-CABLES 1" DIAMETER	16-CABLES 1⅛" DIAMETER 6X37-STRANDING-SPECIAL FLEXIBLE MONITOR PLOW STEEL
QUESTION No.5 LENGTH OF CABLES	420'-0" EACH	382'-0" 6X37 STRANDING	4 LENGTHS 381'-0" EACH 6X37 STRANDING	340'-0" 360'-0"	2 LENGTHS 350'-0" EACH
QUESTION No.6 ARE SAFETY CABLES USED	SAFETY EQUALIZER BEAM WITH SHOCK SPRINGS		NO	YES	NO
QUESTION No.7 DO CABLES HAVE HEMP OR IRON CORD	HEMP	STEEL CENTER	HEMP	HEMP	HEMP
QUESTION No.8 DIAMETER OF MAIN SHEAVES	36" 34"	33¾"	33¾"	30"	34" 36"
QUESTION No.9 DIAMETER OF EQUALIZER SHEAVES	NONE	33¾"	33¾"	30"	NONE
QUESTION No.10 DIAMETER OF MAIN HOIST DRUM	49⅝"	38⅛"	38"	48"	36"
QUESTION No.11 ARE HOOKS SOLID OR LAMINATED	LAMINATED	LAMINATED	LAMINATED	LAMINATED	LAMINATED
QUESTION No.12 INSPECTION OF CABLES (A) WHEN INSPECTED (B) BY WHOM INSPECTED (C) HOW INSPECTED	ONCE EACH WEEK BY CABLE INSPECTOR AND ONCE EACH MONTH JOINTLY BY CABLE INSPECTOR AND MILLWRIGHT FOREMAN. CABLES ARE WASHED OFF WITH TORCH OIL FOR A SPACE APPROX. 18" WIDE ACROSS THE DRUM EACH TIME IT IS INSPECTED AND THEN EACH CABLE IS INSPECTED OVER IT'S ENTIRE LENGTH AS IT IS WOUND OFF THE DRUM	EVERY DAY & WEEK ENDS BY CRANE INSPECTOR HOOK LOWERED SO CABLE CAN BE INSPECTED	EVERY WEEK CABLE INSPECTOR RIDES THE HOOKS	ONCE A WEEK CABLE SPLICER INSPECTOR RIDES BAIL	ONCE A WEEK BY FOREMAN OR HEAD MOTOR INSPECTAR WITH HOOKS LOWERED TO GROUND ALSO RAISED TO FULL HEIGHT AND CABLE ON DRUM INSPECTED WEAR ON OUTER WIRES IS NOT PERMITTED TO REDUCE THE DIAMETER OF THESE OUTER WIRES TO LESS THAN ½ THEIR ORIGINAL DIAMETER
QUESTION No.13 IS TEST LIFT MADE TO ASCERTAIN IF BRAKES ARE HOLDING BEFORE TAKING HEAT AWAY FROM STAND	YES	YES	YES	YES	YES
QUESTION No.14 AVERAGE LIFE OF CABLES ON THIS CRANE	CRANE IN SERVICE 1½ YEARS NO CHANGE TO DATE	12 MONTHS	3 MONTHS FULL OPERATIONS	8 MONTHS	12 MONTHS
QUESTION No.15 MAXIMUM LENGTH OF TIME CABLES ARE LEFT IN SERVICE ON THIS CRANE		12 MONTHS	12 MONTHS PRESENT RATE OF OPERATIONS	10 TO 12 MONTHS	12 MONTHS
QUESTION No.16 TYPE OF LUBRICANT USED ON CABLES AND HOW IS IT APPLIED	CYLINDER OIL IS POURED ON CABLES THROUGH SPOUT AS LADLE BEAM IS BEING LOWERED & RAISED	GRAPHITE AND CYLINDER OIL APPLIED WITH A PAINT BRUSH	IRONSIDE GEAR SHIELD SWAB	BLACK OIL AND GRAPHITE	BLACK OIL APPLIED ON CABLE DRUM AND LOWER SHEAVES
QUESTION No.17 DOES CRANE HAVE TO HOLD LADLE WHILE HEAT IS BEING TAPPED OR WHILE ADDITIONS ARE BEING MADE IN LADLE	NO	YES	NO	NO	NO
QUESTION No.18 REMARKS	CABLE LIFE FOR THIS TYPE CRANE STATED BY CRANE MFG. SHOULD BE 2 TO 3 YEARS		CABLES ARE CHANGED ONE AT A TIME		THIS CRANE WAS DESIGNED FOR 150 TONS. CRANE RECEIVES CLOSE INSPECTION AND ATTENTION

	PLANT NO. 21	PLANT NO. 22	PLANT NO. 23	PLANT NO. 24	PLANT NO. 25
QUESTION No. 1 LOCATION OF CRANE	OPEN HEARTH PITS	OPEN HEARTH PITS	OPEN HEARTH PITS	OPEN HEARTH PITS	OPEN HEARTH PITS
QUESTION No. 2 CAPACITY OF CRANE	125 TONS	125 TONS	125 TONS	125 TONS	125 TONS
QUESTION No. 3 MAXIMUM LIFT OF CRANE IN ACTIVE OPERATION					
(A) WEIGHT OF HOT METAL					210,000-LBS.
(B) WEIGHT OF HOOKS					12,000-LBS.
(C) WEIGHT OF SPREADER BEAM					16,000-LBS.
(D) WEIGHT OF CABLES					2,304-LBS.
(E) WEIGHT OF LADLE & LINING					87,000-LBS.
(F) TOTAL WEIGHT	240,000-LBS.	240,000-LBS.	300,000-LBS.	300,000-LBS.	327,804-LBS.
QUESTION No. 4 SIZE AND NUMBER OF CABLES	2-CABLES 1⅛" DIAMETER	2-CABLES 1" DIAMETER	2-CABLES 1⅛" DIAMETER	2-CABLES 1" DIAMETER	4-CABLES 1" DIAMETER 6X19 STRANDING
QUESTION No. 5 LENGTH OF CABLES			395'-0" EACH	545'-0" EACH	360'-0" EACH
QUESTION No. 6 ARE SAFETY CABLES USED					YES BUT NOT USED AS SUCH
QUESTION No. 7 DO CABLES HAVE HEMP OR IRON CORD	STEEL CENTER	STEEL CENTER	STEEL CENTER	STEEL CENTER	HEMP
QUESTION No. 8 DIAMETER OF MAIN SHEAVES	30"	28"	30"	26¼"	30"
QUESTION No. 9 DIAMETER OF EQUALIZER SHEAVES		28"	30"	26¼"	15" 4½"
QUESTION No. 10 DIAMETER OF MAIN HOIST DRUM	38"	30"	38"	29⅞"	43"
QUESTION No. 11 ARE HOOKS SOLID OR LAMINATED	LAMINATED	LAMINATED	LAMINATED	LAMINATED	LAMINATED
QUESTION No. 12 INSPECTION OF CABLES					
(A) WHEN INSPECTED	ONCE A WEEK	ONCE A WEEK			EVERY DAY
(B) BY WHOM INSPECTED	BY CRANE INSPECTOR AND TURN FOREMAN	BY CRANE INSPECTOR AND TURN FOREMAN	BY CRANE INSPECTOR	BY CRANE INSPECTOR	BY FOREMAN OF THIS DEPARTMENT
(C) HOW INSPECTED					WITH HEAVY GLOVE AND PIECE OF WASTE
QUESTION No. 13 IS TEST LIFT MADE TO ASCERTAIN IF BRAKES ARE HOLDING BEFORE TAKING HEAT AWAY FROM STAND	YES	YES	YES	YES	YES
QUESTION No. 14 AVERAGE LIFE OF CABLES ON THIS CRANE	6 MONTHS	6 MONTHS	6 MONTHS	6 MONTHS	4 MONTHS
QUESTION No. 15 MAXIMUM LENGTH OF TIME CABLES ARE LEFT IN SERVICE ON THIS CRANE	12 MONTHS	12 MONTHS	12 MONTHS	12 MONTHS	5 MONTHS
QUESTION No. 16 TYPE OF LUBRICANT USED ON CABLES AND HOW IS IT APPLIED	BLACK OIL ON DRUM	BLACK OIL ON DRUM	SPECIAL CABLE COMPOUND	SPECIAL CABLE COMPOUND	BLACK OIL PUT ON WITH PAINT BRUSH
QUESTION No. 17 DOES CRANE HAVE TO HOLD LADLE WHILE HEAT IS BEING TAPPED OR WHILE ADDITIONS ARE BEING MADE IN LADLE	NO	NO	NO	NO	AFTER LADLE IS 3/4 FULL CRANE TAKES HOLD OF LADE AND HOLDS UNTIL FULL
QUESTION No. 18 REMARKS	CABLE INSPECTED BY CRANE INSPECTOR AND TURN FOREMAN ON DIFFERENT DAYS OF EACH WEEK AND WRITTEN REPORT MADE	CABLE INSPECTED BY CRANE INSPECTOR AND TURN FOREMAN ON DIFFERENT DAYS OF EACH WEEK AND WRITTEN REPORT MADE			AUX.-1 PC. 7/8" CABLE 300'-0" LONG

	PLANT NO.26	PLANT NO.27	PLANT NO.28	PLANT NO.29	PLANT NO.30
QUESTION No.1 LOCATION OF CRANE	POURING SIDE OPEN HEARTH	OPEN HEARTH	OPEN HEARTH PITS	OPEN HEARTH PITS	OPEN HEARTH
QUESTION No.2 CAPACITY OF CRANE	150 TONS	150 TONS	150 TONS	160 TONS	165 TONS
QUESTION No.3 MAXIMUM LIFT OF CRANE IN ACTIVE OPERATION					
(A) WEIGHT OF HOT METAL	280,000-LBS.	250,000-LBS.	220,000-LBS.	224,000-LBS.	235,000-LBS.
(B) WEIGHT OF HOOKS	15,500-LBS.	5,000-LBS.	11,200-LBS.	10,000-LBS.	11,936-LBS.
(C) WEIGHT OF SPREADER BEAM	12,000-LBS.	7,000-LBS.	5,650-LBS.	7,000-LBS.	29,950-LBS.
(D) WEIGHT OF CABLES	1,700-LBS.	3,000-LBS.	2,350-LBS.	2,800-LBS.	4,280-LBS.
(E) WEIGHT OF LADLE & LINING	80,000-LBS.	84,000-LBS.	100,000-LBS.	86,000-LBS.	89,600-LBS.
(F) TOTAL WEIGHT	389,200-LBS.	349,000-LBS.	339,200-LBS.	329,800-LBS.	370,766-LBS.
QUESTION No.4 SIZE AND NUMBER OF CABLES	2-CABLES 1" DIAMETER	4-CABLES 1⅛" DIAMETER	4-CABLES 1⅛" DIAMETER	4-CABLES 1⅛" DIAMETER SPECIAL FLEXIBLE MONITOR 6X37 STRANDING	44-CABLES 1⅛" DIAMETER
QUESTION No.5 LENGTH OF CABLES	550'-0" EACH	305'-0" EACH	270'-0" EACH	350'-0" EACH	4-LENGTHS 535'-0" EACH
QUESTION No.6 ARE SAFETY CABLES USED	NO	NO	NO	YES	YES
QUESTION No.7 DO CABLES HAVE HEMP OR IRON CORD	HEMP	SOFT IRON	INDEPENDENT WIRE ROPE CENTER	7X7 SOFT IRON CENTER	HEMP
QUESTION No.8 DIAMETER OF MAIN SHEAVES	26¼"	37½"	37½"	37½"	33¾"
QUESTION No.9 DIAMETER OF EQUALIZER SHEAVES	26¼"	33¾"	33¾"	33¾"	NONE
QUESTION No.10 DIAMETER OF MAIN HOIST DRUM	30"	51"	51"	50⅞"	66½"
QUESTION No.11 ARE HOOKS SOLID OR LAMINATED	LAMINATED	LAMINATED	LAMINATED	LAMINATED	LAMINATED
QUESTION No.12 INSPECTION OF CABLES (A) WHEN INSPECTED (B) BY WHOM INSPECTED (C) HOW INSPECTED		ONCE A WEEK BY LOWERING BEAM DOWN AND LOOKING OVER CABLE FOR WEAR OR BREAKS IN STRANDS BY FOREMAN IN CHARGE	ONCE A WEEK BY SIGHT OCCASIONALLY CHECKING GROOVED DIAMETER OF SHEAVES AND CHECK CLAMPS ON DEAD END OF CABLES. GENERAL FOREMAN AND MOTOR INSPECTOR	ONCE A WEEK BY CRANE MECHANIC RIDING SPREADER AND USING HEAVY GLOVES ONCE A MONTH BY RIGGERS RIDING SPREADER AND USING WASTE TO DETECT BROKEN WIRES	ONCE A WEEK BY O.H. ONCE A MONTH BY RIGGER PERSONAL INSPECTION AS CABLE PASSES A POINT
QUESTION No.13 IS TEST LIFT MADE TO ASCERTAIN IF BRAKES ARE HOLDING BEFORE TAKING HEAT AWAY FROM STAND	YES	YES	YES	YES	YES
QUESTION No.14 AVERAGE LIFE OF CABLES ON THIS CRANE	12 MONTHS	12 MONTHS NORMAL TIMES	12 MONTHS	18 MONTHS	12 MONTHS
QUESTION No.15 MAXIMUM LENGTH OF TIME CABLES ARE LEFT IN SERVICE ON THIS CRANE	24 MONTHS	12 MONTHS NORMAL TIMES	TO YEAR 1930 12 MONTHS SINCE 1930 ON INSPECTION	NO SET TIME	12 MONTHS
QUESTION No.16 TYPE OF LUBRICANT USED ON CABLES AND HOW IS IT APPLIED		CAR OIL POURED ON FROM A CAN	KEYSTONE VELOX CABLE GREASE APPLIED WITH A PAINT BRUSH	STANDARD OIL CO. JOURNAL COMPOUND APPLIED TO MOVING CABLE FROM SPREADER AND GROOVES OF HOISTING DRUM	EXCESS GREASE FROM SHEAVES
QUESTION No.17 DOES CRANE HAVE TO HOLD LADLE WHILE HEAT IS BEING TAPPED OR WHILE ADDITIONS ARE BEING MADE IN LADLE	NO	NOT GENERAL PRACTICE-SOME TIMES WHEN LADLE STOOL IS NOT IN GOOD REPAIR	YES	YES	NO
QUESTION No.18 REMARKS				WHEN A CABLE HAS BEEN IN SERVICE 3 MONTHS CRANE MECHANIC CHECKS DIAMETER OF CABLE FOR STRETCHING	

	PLANT NO.31	PLANT NO.32	PLANT NO.33	PLANT NO.34	PLANT NO.35
QUESTION No.1 LOCATION OF CRANE	OPEN HEARTH	POURING SIDE OPEN HEARTH	POURING SIDE OPEN HEARTH	OPEN HEARTH	POURING SIDE OPEN HEARTH
QUESTION No.2 CAPACITY OF CRANE	165 TONS	175 TONS	175 TONS	175 TONS	175 TONS
QUESTION No.3 MAXIMUM LIFT OF CRANE IN ACTIVE OPERATION					
(A) WEIGHT OF HOT METAL	235,000-LBS.	313,600-LBS.	280,000-LBS.	284,000-LBS.	220,000-LBS.
(B) WEIGHT OF HOOKS	13,570-LBS.	28,500-LBS.	15,500-LBS.	7,000-LBS.	13,600-LBS.
(C) WEIGHT OF SPREADER BEAM	17,330-LBS.		16,400-LBS.	21,000-LBS.	14,500-LBS.
(D) WEIGHT OF CABLES	2,340-LBS.		3,600-LBS.	4,000-LBS.	3,000-LBS.
(E) WEIGHT OF LADLE & LINING	89,600-LBS.	93,135-LBS.	80,000-LBS.	76,000-LBS.	85,000-LBS.
(F) TOTAL WEIGHT	357,840-LBS.	435,235-LBS.	395,500-LBS.	392,000-LBS.	336,100-LBS.
QUESTION No.4 SIZE AND NUMBER OF CABLES	24-CABLES 1⅛" DIAMETER	4-CABLES 1⅛" DIAMETER	8-CABLES 1⅛" DIAMETER "4 SAFETY CABLES"	4-CABLES 1⅛" DIAMETER	36-CABLES 1⅛" DIAMETER
QUESTION No.5 LENGTH OF CABLES	2 LENGTHS 560'-0" EACH	2 LENGTHS 570'-0" EACH 2 LENGTHS 576'-0" EACH	225'-0" EACH	563'-0"	4 LENGTHS 405'-0" EACH
QUESTION No.6 ARE SAFETY CABLES USED	NO	YES	YES	NO	NO
QUESTION No.7 DO CABLES HAVE HEMP OR IRON CORD	HEMP	HEMP	HEMP	HEMP	SOFT IRON
QUESTION No.8 DIAMETER OF MAIN SHEAVES	33¾"	34"	34"	32½"	36"
QUESTION No.9 DIAMETER OF EQUALIZER SHEAVES	12"	18" 10½"	32"	NONE	36"
QUESTION No.10 DIAMETER OF MAIN HOIST DRUM	37½"	50"	50⅛"	60½"	50"
QUESTION No.11 ARE HOOKS SOLID OR LAMINATED	LAMINATED	LAMINATED	LAMINATED	LAMINATED	LAMINATED
QUESTION No.12 INSPECTION OF CABLES (A) WHEN INSPECTED (B) BY WHOM INSPECTED (C) HOW INSPECTED	ONCE A WEEK BY O.H. ONCE A MONTH BY RIGGERS PERSONAL INSPECTION AS CABLE PASSES A POINT	EACH DAY LADLE CRANE INSPECTOR RIDING SPREADER		WEEKLY BY CRANE INSPECTOR ON DRUMS AND SPREADER ALSO WATCHED DAILY BY RIGGERS AND CRANE OPERATOR	EVERY DAY MILLWRIGHT VISUAL INSPECTION
QUESTION No.13 IS TEST LIFT MADE TO ASCERTAIN IF BRAKES ARE HOLDING BEFORE TAKING HEAT AWAY FROM STAND	YES	YES	YES	NO-NOT ENOUGH HEADROOM-TEST IS MADE AS SOON AS LADLE LEAVES STAND IN CENTER OF PIT	YES
QUESTION No.14 AVERAGE LIFE OF CABLES ON THIS CRANE	6 MONTHS	8 MONTHS	24 MONTHS	13 MONTHS	18 MONTHS
QUESTION No.15 MAXIMUM LENGTH OF TIME CABLES ARE LEFT IN SERVICE ON THIS CRANE	6 MONTHS	12 MONTHS	24 MONTHS	15 MONTHS	24 MONTHS
QUESTION No.16 TYPE OF LUBRICANT USED ON CABLES AND HOW IS IT APPLIED	EXCESS GREASE FROM SHEAVES	STANDARD OIL CABLE GREASE POURED INTO DRUM GROOVES AS CABLES ARE COILED ON		BLACK OIL OR USED CYLINDER OIL AND ENGINE OIL MIXED. APPLIED WITH A BRUSH	TAR AND BLACK OIL HEATED AND MIXED. POURED ON CABLES WHILE HOIST IS IN MOTION
QUESTION No.17 DOES CRANE HAVE TO HOLD LADLE WHILE HEAT IS BEING TAPPED OR WHILE ADDITIONS ARE BEING MADE IN LADLE	NO	NO	NO	NO-BUT GENERALLY HOOK ON WHEN LADLE IS ⅔ FULL	NO-HOOKS ARE PLACED UNDER TRUNIONS & LOCKED WHEN LADLE IS ABOUT ¼ FULL TO PREVENT INTERFERENCE FROM OVERFLOWING SLAG
QUESTION No.18 REMARKS					

	PLANT NO.36	PLANT NO.37	PLANT NO.38	PLANT NO.39	PLANT NO.40
QUESTION No.1 LOCATION OF CRANE	OPEN HEARTH	PIT SIDE OPEN HEARTH			POURING SIDE OPEN HEARTH
QUESTION No.2 CAPACITY OF CRANE	175 TONS	175 TONS	175 TONS	175 TONS	175 TONS
QUESTION No.3 MAXIMUM LIFT OF CRANE IN ACTIVE OPERATION					
(A) WEIGHT OF HOT METAL	275,000-LBS.	296,000-LBS.	240,000-LBS.	260,000-LBS.	313,600-LBS.
(B) WEIGHT OF HOOKS	16,800-LBS.	12,750-LBS.	14,000-LBS.	11,400-LBS.	9,000-LBS.
(C) WEIGHT OF SPREADER BEAM	33,000-LBS.	17,000-LBS.	30,400-LBS.	24,000-LBS.	21,000-LBS.
(D) WEIGHT OF CABLES	3,100-LBS.	4000-LBS.	3,900-LBS.	3,248-LBS.	4,000-LBS.
(E) WEIGHT OF LADLE & LINING	106,000-LBS.	102,000-LBS.	87,500-LBS.	83,000-LBS.	118,000-LBS.
(F) TOTAL WEIGHT	433,900-LBS.	431,750-LBS.	375,800-LBS.	381,648-LBS.	465,600-LBS.
QUESTION No.4 SIZE AND NUMBER OF CABLES	36-CABLES 1⅛" DIAMETER	36-CABLES 1⅛" DIAMETER	36-CABLES 1⅛" DIAMETER	4-CABLES 1⅛" DIAMETER	4-CABLES 1⅛" DIAMETER 6X37 STRANDING
QUESTION No.5 LENGTH OF CABLES	395'-0"	4 LENGTHS 502'-0" EACH	4 LENGTHS 485'-0" EACH	403'-0"	475'-0" EACH
QUESTION No.6 ARE SAFETY CABLES USED	DOUBLE REEVED	NO	NO	ONE CABLE CAN BREAK AND OTHERS WILL TAKE CARE OF IT	NO
QUESTION No.7 DO CABLES HAVE HEMP OR IRON CORD	IRON	HEMP	IRON	IRON	IRON
QUESTION No.8 DIAMETER OF MAIN SHEAVES	34" 36"	33¾" 37½"	33¾" 36"	33¾"	34" 32"
QUESTION No.9 DIAMETER OF EQUALIZER SHEAVES	EQUALIZER BAR	DEAD END TO EQUALIZER BAR ON SPREADER	NONE-CABLE DEAD ENDED ON LIFTING BEAM	37½"	CABLE DEAD ENDED
QUESTION No.10 DIAMETER OF MAIN HOIST DRUM	49⅝"	58"	54"	60"	49⅝"
QUESTION No.11 ARE HOOKS SOLID OR LAMINATED	LAMINATED	LAMINATED	LAMINATED	LAMINATED	LAMINATED
QUESTION No.12 INSPECTION OF CABLES (A) WHEN INSPECTED (B) BY WHOM INSPECTED (C) HOW INSPECTED	WEEKLY BY FOREMAN AND MOTOR INSPECTOR-INSPECT ALL PARTS LIABLE TO WEAR	MONTHLY BY FOREMAN AND MOTOR INSPECTOR VISUAL AND TOUCH	DAILY BY CRANEMAN, WEEKLY BY FOREMAN & HEAD MOTOR INSPECTOR INSPECTED FOR UNDUE WEAR OR BROKEN WIRES WITH LOWERED TO GROUND ALSO WHEN MOST OF CABLE IS WRAPPED ON DRUM	EVERY 24 HOURS TURN LEADER LOOKING OVER THEM	EACH WEEK PLANT SAFETY INSPECTOR RIDES THE HEAD UP & DOWN HOLDING HIS HANDS OVER THE CABLES WITH A PIECE OF WASTE IN EACH HAND LOOKING FOR BROKEN WIRES WHICH WILL BE READILY FELT
QUESTION No.13 IS TEST LIFT MADE TO ASCERTAIN IF BRAKES ARE HOLDING BEFORE TAKING HEAT AWAY FROM STAND	YES	YES	YES	YES	YES-LIFTS 6" FROM STAND THEN STOPS TO TEST BRAKES
QUESTION No.14 AVERAGE LIFE OF CABLES ON THIS CRANE	14 MONTHS	DON'T KNOW	UNKNOWN	12 MONTHS	12 MONTHS
QUESTION No.15 MAXIMUM LENGTH OF TIME CABLES ARE LEFT IN SERVICE ON THIS CRANE	NO FIXED MAXIMUM	ONCE EACH YEAR ON 100% OPERATION	UNKNOWN	250,000 TONS	UNTIL OUTSIDE WIRES ARE WORM FLAT OR START TO BREAK
QUESTION No.16 TYPE OF LUBRICANT USED ON CABLES AND HOW IS IT APPLIED	RESIDIUM OIL PAINTED ON WITH BRUSH	CRATER COMPOUND POURED ON DRUM AND SHEAVES	WINTER-BLACK OIL APPLIED TO DRUM, SHEAVES AND HOOK BLOCK	OIL USED POURED ON FROM OIL CAN	A PLUMBAGO GREASE COMPOUND IS USED AND PUT ON BY HAND
QUESTION No.17 DOES CRANE HAVE TO HOLD LADLE WHILE HEAT IS BEING TAPPED OR WHILE ADDITIONS ARE BEING MADE IN LADLE	NO	NO	ORDINARILY NO ON RARE OCCASIONS IT IS NECESSARY WHEN REPAIRS ARE BEING MADE TO STANDS	NO	NO
QUESTION No.18 REMARKS			CRANE INSTALLED SIX MONTHS AGO AND NO DATA IS AVAILABLE AS YET REGARDING CABLE LIFE		CABLE LIFE CAN HARDLY BE JUDGED BY TIME. IT SHOULD BE JUDGED BY TONS HANDLED

	PLANT NO. 41	PLANT NO. 42	PLANT NO. 43	PLANT NO. 44	PLANT NO. 45
QUESTION No. 1 LOCATION OF CRANE	POURING BAY OPEN HEARTH	OPEN HEARTH	POURING SIDE OPEN HEARTH	OPEN HEARTH	OPEN HEARTH
QUESTION No. 2 CAPACITY OF CRANE	200 TONS	200 TONS	200 TONS	200 TONS	200 GROSS TONS
QUESTION No. 3 MAXIMUM LIFT OF CRANE IN ACTIVE OPERATION					
(A) WEIGHT OF HOT METAL	400,000-LBS.	220,000-LBS.	300,000-LBS.	336,000-LBS.	358,400-LBS.
(B) WEIGHT OF HOOKS	22,400-LBS.	28,000-LBS.	22,000-LBS.	26,000-LBS.	22,800-LBS.
(C) WEIGHT OF SPREADER BEAM	35,500-LBS.	20,000-LBS.	35,000-LBS.	30,000-LBS.	42,400-LBS.
(D) WEIGHT OF CABLES	5,280-LBS.	20,000-LBS.	4,000-LBS.	5,200-LBS.	8,300-LBS.
(E) WEIGHT OF LADLE & LINING	136,000-LBS.	70,000-LBS.	107,000-LBS.	120,000-LBS.	128,260-LBS.
(F) TOTAL WEIGHT	599,180-LBS.	358,000-LBS.	468,000-LBS.	517,200-LBS.	560,160-LBS.
QUESTION No. 4 SIZE AND NUMBER OF CABLES	4-CABLES 1⅛" DIAMETER REEVED IN 44 PARTS	44-CABLES 1⅛" DIAMETER	4-CABLES 1⅛" DIAMETER	4-CABLES 1⅛" DIAMETER 44 LENGTHS BY REEVING MONITOR PLOW STEEL	52-CABLES 1¼" DIAMETER
QUESTION No. 5 LENGTH OF CABLES	660'-0" EACH	4 LENGTHS 625'-0" EACH	600'-0" EACH	590'-0"	4 LENGTHS 860'-0" EACH
QUESTION No. 6 ARE SAFETY CABLES USED	SAFETY EQUALIZER BEAM WITH SHOCK SPRINGS	NO	YES	2-CABLES WILL CARRY LOAD	YES
QUESTION No. 7 DO CABLES HAVE HEMP OR IRON CORD	HEMP	STEEL	HEMP	IRON	METALLIC
QUESTION No. 8 DIAMETER OF MAIN SHEAVES	36" 34"	33¾"	37½"	33¾" 37½"	37½" 41¼"
QUESTION No. 9 DIAMETER OF EQUALIZER SHEAVES	NONE	NONE	33¾"	USE EQUALIZER BEAMS. 2 ON SPREADER	NONE
QUESTION No. 10 DIAMETER OF MAIN HOIST DRUM	57"	63"	63"	63"	63"
QUESTION No. 11 ARE HOOKS SOLID OR LAMINATED	LAMINATED	LAMINATED	LAMINATED	LAMINATED	LAMINATED
QUESTION No. 12 INSPECTION OF CABLES (A) WHEN INSPECTED (B) BY WHOM INSPECTED (C) HOW INSPECTED	ONCE EACH WEEK BY CABLE INSPECTOR AND ONCE EACH MONTH JOINTLY BY CABLE INSPECTOR AND MILLWRIGHT FOREMAN. CABLES ARE WASHED OFF WITH TORCH OIL FOR A SPACE APPROX. 12" WIDE ACROSS THE DRUM EACH TIME IT IS INSPECTED AND THEN EACH CABLE IS INSPECTED OVER IT'S ENTIRE LENGTH AS IT IS WOUND OFF THE DRUM	MONTHLY BY ASSISTANT ELECTRICAL FOREMAN VISUAL INSPECTION	DAILY GENERAL WEEKLY RIGID. MAN RIDES BEAM CLEANS CABLE AS IT PASSES ELECTRICIAN AND MILLWRIGHT REGULARLY CHECKED BY FOREMAN	WEEKLY-EACH SUNDAY MORNING BY RIDING SPREADER BEAM INSPECTION BY RIGGERS	NO ANSWER
QUESTION No. 13 IS TEST LIFT MADE TO ASCERTAIN IF BRAKES ARE HOLDING BEFORE TAKING HEAT AWAY FROM STAND	YES	ALWAYS	YES	YES	NO ANSWER
QUESTION No. 14 AVERAGE LIFE OF CABLES ON THIS CRANE	CRANE IN SERVICE 1½ YEARS NO CHANGE TO DATE	18 MONTHS ESTIMATED	24 MONTHS	12 MONTHS	NO ANSWER
QUESTION No. 15 MAXIMUM LENGTH OF TIME CABLES ARE LEFT IN SERVICE ON THIS CRANE		DEPENDS ON INSPECTION	24 MONTHS	12 MONTHS	NO ANSWER
QUESTION No. 16 TYPE OF LUBRICANT USED ON CABLES AND HOW IS IT APPLIED	CYLINDER OIL IS POURED ON CABLES THROUGH SPOUT AS LADLE BEAM IS BEING LOWERED AND RAISED	CYLINDER OIL POURED ON DRUM AND BRUSHED ON CABLES	CABLE SHIELD WR-4L IRONSIDES HEATED AND BRUSHED ON	BLACK OIL APPLIED WITH BRUSH. ALSO USE CABLE SHIELD	NO ANSWER
QUESTION No. 17 DOES CRANE HAVE TO HOLD LADLE WHILE HEAT IS BEING TAPPED OR WHILE ADDITIONS ARE BEING MADE IN LADLE	NO	NO-ONLY HELD FOR DRAINAGE	NO	NO	NO ANSWER
QUESTION No. 18 REMARKS	CABLE LIFE FOR THIS TYPE CRANE STATED BY CRANE MFG. SHOULD BE 2 TO 3 YEARS	BRIDGE SPEED OF THIS CRANE REDUCED FROM 520' PER MIN. TO 310' PER MIN.	CAREFUL FIELD AND SHOP INSPECTION OF SHEAVES GROOVES TO MAINTAIN PROPER GROOVE SIZES		CRANES HAVE NOT BEEN PLACED IN OPERATION

ADVERTISING PAGES REMOVED

	PLANT NO. 46	PLANT NO. 47	PLANT NO. 48	PLANT NO. 49	
QUESTION No. 1 LOCATION OF CRANE		POURING SIDE OPEN HEARTH	POURING SIDE OPEN HEARTH	POURING SIDE OPEN HEARTH	
QUESTION No. 2 CAPACITY OF CRANE	225 TONS	250 TONS	250 TONS	275 TONS	
QUESTION No. 3 MAXIMUM LIFT OF CRANE IN ACTIVE OPERATION					
(A) WEIGHT OF HOT METAL	292,000-LBS.	400,000-LBS.	254,000-LBS.	336,000-LBS.	
(B) WEIGHT OF HOOKS	18,130-LBS.	28,000-LBS	17,000-LBS.	12,000-LBS.	
(C) WEIGHT OF SPREADER BEAM	34,000-LBS.	10,000-LBS.	53,000-LBS.	51,000-LBS.	
(D) WEIGHT OF CABLES	4,640-LBS.	6,280-LBS.	5,210-LBS.	6,450-LBS.	
(E) WEIGHT OF LADLE & LINING	90,000-LBS.	150,000-LBS.	148,000-LBS.	114,000-LBS.	
(F) TOTAL WEIGHT	438,770-LBS.	594,280-LBS.	477,210-LBS.	531,450-LBS.	
QUESTION No. 4 SIZE AND NUMBER OF CABLES	4-CABLES 1⅛" DIAMETER	4-CABLES 1⅛" DIAMETER 48 LINES	7" STYLE "B" PATENT FLAT STRAND	4-CABLES 1¼" DIAMETER 6X37 STRANDING	
QUESTION No. 5 LENGTH OF CABLES	580'-0"	785'-0" EACH	4 LENGTHS 620'-0" EACH	665'-0" EACH	
QUESTION No. 6 ARE SAFETY CABLES USED	ONE CABLE CAN BREAK AND THE OTHERS WILL TAKE CARE OF IT	NO	NO	NO	
QUESTION No. 7 DO CABLES HAVE HEMP OR IRON CORD	IRON	INDEPENDENT BLUE STEEL CENTER	IRON	HEMP	
QUESTION No. 8 DIAMETER OF MAIN SHEAVES.	33¾"	37"	40" 42"	37½"	
QUESTION No. 9 DIAMETER OF EQUALIZER SHEAVES	37½"	BAR	NONE	NONE	
QUESTION No. 10 DIAMETER OF MAIN HOIST DRUM	60"	63"	56⅝"	60"	
QUESTION No. 11 ARE HOOKS SOLID OR LAMINATED	LAMINATED	LAMINATED	LAMINATED	LAMINATED	
QUESTION No. 12 INSPECTION OF CABLES (A) WHEN INSPECTED (B) BY WHOM INSPECTED (C) HOW INSPECTED	EVERY 24 HOURS TURN FOREMAN LOOKING THEM OVER	DAILY-VISUAL DAILY AND RAG HELD ON CABLE EVERY WEEK TO DETECT BROKEN STRANDS. DAILY AFTER FIRST BROKEN IS DETECTED BY CRANE MECHANICS	EVERY MONTH BY GETTING ON SPREADER INSPECTED BY PLANT CABLE INSPECTOR	DAILY BY MOTOR INSPECTOR AND WEEKLY BY TURN FOREMAN	
QUESTION No. 13 IS TEST LIFT MADE TO ASCERTAIN IF BRAKES ARE HOLDING BEFORE TAKING HEAT AWAY FROM STAND	YES	YES	YES	YES	
QUESTION No. 14 AVERAGE LIFE OF CABLES ON THIS CRANE	12 MONTHS	10 MONTHS	12 MONTHS	NOT YET DETERMINED	
QUESTION No. 15 MAXIMUM LENGTH OF TIME CABLES ARE LEFT IN SERVICE ON THIS CRANE	400,000 TONS	15 MONTHS DEPENDS ON OPERATIONS	12 MONTHS	NOT KNOWN	
QUESTION No. 16 TYPE OF LUBRICANT USED ON CABLES AND HOW IS IT APPLIED	OIL USED POURED ON FROM OIL CAN	USE MOTOR OIL AND APPLIED WITH WHITE WASH BRUSH	PLUMBAGO NO. 3 APPLIED WITH A PADDLE	STANDARD OIL CO. CABLE DRESSING APPLIED BY POURING ON WHILE CABLE RUNS OVER SHEAVES	
QUESTION No. 17 DOES CRANE HAVE TO HOLD LADLE WHILE HEAT IS BEING TAPPED OR WHILE ADDITIONS ARE BEING MADE IN LADLE	NO	YES	NO	NO	
QUESTION No. 18 REMARKS		AVERAGE NUMBER OF HEATS HANDLED - 660			